ENGINEERING

G. Croydon Marks

NIAGARA FALLS. [*Frontispiece.*

HYDRAULIC POWER ENGINEERING

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HYDRAULIC POWER ENGINEERING

A PRACTICAL MANUAL

ON THE CONCENTRATION AND TRANSMISSION OF

POWER BY HYDRAULIC MACHINERY

BY

G. CROYDON MARKS

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FELLOW OF THE CHARTERED INSTITUTE OF PATENT AGENTS

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PREFACE.

THIS work may be regarded as a successor to a smaller volume by the same Author on "HYDRAULIC MACHINERY," published in 1891, which he prepared with a view to the assistance of engineering students and others who might be practically interested in the subject.

In the present volume an attempt is made to give an outline discussion and description of the main points and principles requiring attention by engineers having the responsibility of designing or constructing works and appliances for the utilisation of water for the transmission of power.

It would be impossible in any single volume to deal adequately or comprehensively with the many problems arising in the different sections into which the very large subject of Hydraulics and Hydraulic Engineering naturally divides itself. The Author, therefore, has contented himself with giving examples which have special reference to the particular sections in which they occur; and in addition, he has endeavoured to lead up to the general subject by a brief examination of the principles underlying the whole study.

The development of hydraulic power machinery has been somewhat of a modern movement, but the examples which are to be found in the following pages

will possibly lead the engineer and designer to go still further in the realisation of the most convenient form of power transmission available for industrial undertakings and commercial manufactures. In practice it is constantly found that new problems are raised, and new forms of machinery required for their satisfactory solution.

The Author wishes to acknowledge here the services which have been rendered him by Mr E. B. Fenby and Mr A. Suggate, members of his staff, in the preparation of the examples and drawings given in the volume, and in compiling many of the tables now published here for the first time. Free use (it should also be mentioned) has been made of information published in the "Proceedings of the Institution of Civil Engineers," the permission of the Council of that Institution having been kindly given for that purpose; and additional information has been obtained from descriptions of works appearing in the *Engineer* and *Engineering*, and in *Cassier's Magazine*. Many of the illustrations have been specially prepared from information kindly placed at the service of the Author by the several engineering firms referred to in the work.

The Author would refer students who may seek fuller information on the question of Hydraulic Motors and Turbines to Mr Bodmer's treatise upon that subject.

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A.

HYDRAULIC POWER ENGINEERING.

CHAPTER I.

PRINCIPLES OF HYDRAULICS.

General Properties of Water.—There are certain properties of water which render it particularly suitable to the requirements of the hydraulic power engineer. There are three distinct methods of using water for transmitting power. By the first method the water is placed at some height above a given datum level, and by its descent is caused to turn a water-wheel. In this case the water acts by its large weight and high viscosity; and if either of these two properties were wanting, the water would be of small use for this method. By the second method the water is subjected to great pressure, and is applied to a piston moving in a cylinder. In this case the water acts by its high viscosity and power to withstand a pressure without serious loss by internal friction.

The difference between these two methods is more apparent when it is pointed out that whereas in the first case the entire absence of weight would mean absolute inefficiency to perform work, in the second case the weight of the water

becomes a serious obstacle to its use, and requires special care to be taken in designing certain hydraulic machinery to prevent mishaps.

Although these two methods appear to be antagonistic, there is the third method requiring the water to have all the properties above enumerated. The water is caused to act by its kinetic energy, and is first subjected to a greater or less pressure, and thus caused to acquire a velocity. This velocity is then abstracted in passing through the machine, and the corresponding energy is thus applied to perform work. This is the principle on which turbines are designed.

If water is allowed to flow unconfined it will not come to rest until its upper surface corresponds to a horizontal plane such as the upper surface of a canal at rest. A horizontal surface is not a flat plane, but is curved to the radius of the earth, and may be defined as that surface in which the force of gravity is the same at all points.

From the above definition it is apparent that the weight of any body varies according as it is placed nearer to or further above the level of the sea. Thus a water-wheel placed on a mountain, and consuming say 20 cubic feet of water per second, will not be doing the same number of horse-powers as if it were placed at the sea-level, and consuming the same quantity of water. If, however, the power is expended in lifting, as, for instance, in connection with a vertical mine shaft, this difference of weight is of no importance, as the weights to be lifted have been reduced in a like proportion. When the power is expended in crushing ore, or overcoming certain frictional resistances, the wheel will require more water on the mountain than at sea-level to overcome the same resistance.

The density of water (*i.e.*, its weight compared to that of come body bulk for bulk) is varied either by a change of temperature or by a change of pressure. In determining specific gravities of bodies distilled water at a temperature

of 62° F. and barometric pressure of 30 inches of mercury is taken as the standard, and called unity. The weight of the body to be compared is observed and compared bulk for bulk to this standard. Thus wrought iron has a specific gravity of 7.8, or 1 cubic inch of wrought iron weighs the same as 7.8 cubic inches of distilled water, each taken at the standard temperature and pressure.

The standard weight of water has been fixed by the Board of Trade at 62.2786 lbs. per cubic foot at 62° F. and 30 inches barometric pressure. The greatest density of water as affected by change of temperature is found to correspond to 39.3° F., and at this temperature 1 cubic foot weighs 62.425 lbs.

As regards change of density by alteration of pressure, one atmosphere (14.7 lbs.) of additional pressure is found to cause a reduction of volume of .00005, and consequent increase of weight of .000050002. If this reduction of volume be assumed to increase directly as the pressure applied, the reductions corresponding to the usually employed hydraulic pressures are :—

750 lbs.	($\frac{1}{3}$ ton)	per sq. in.,	.00254 =	4.386	in. per cub. ft.
1,500 ,,	($\frac{2}{3}$ ton)	,,	.00508 =	8.772	,,
2,240 ,,	(1 ton)	,,	.00761 =	13.158	,,
4,480 ,,	(2 tons)	,,	.01522 =	26.316	,,
6,720 ,,	(3 tons)	,,	.02283 =	39.474	,,

Wrought iron subjected to a pressure of 1 ton per square inch is compressed to the extent of .000077 of its length, hence water is about three times as elastic as iron.

The water employed by the hydraulic engineer is either river water or town service water, and in either case foreign substances are carried in solution, thereby altering the density. In unfiltered water small particles of matter are carried in suspension, and as these particles have almost without exception a greater density than the water, a further increase of density is encountered.

The following are the average weights per cubic foot of different samples:—

River water	- - -	62.5 lbs. = 1,000 oz.
Salt water	- - -	64.0 ,,
Dead Sea	- - -	73.0 ,,

At a temperature of 32° F., and barometric pressure of 30 inches, water passes into the solid form called ice, and owing to the great change in viscosity is useless in this form for the purposes of the hydraulic engineer. This change of condition from the liquid to the solid is accompanied by a change of volume and consequent change of density. The weight of 1 cubic foot of ice is 57.5 lbs.

An increase of pressure delays solidification, as also does absolute rest of the particles of water; but as reduction of pressure to atmosphere and agitation both bring about rapid solidification, this property of retarded solidification is of no moment, as in all hydraulic appliances the water is subject to both atmospheric pressure and agitation during the performance of its function.

Hydrostatics.—The name hydrostatics is given to the study of the principles governing the conditions of equilibrium of a column or quantity of water.

Pascal's Principle.—Pascal discovered that if water be enclosed in a vessel and a pressure applied, as for instance by pressing on a piston in a cylinder attached to the vessel, that the pressure is transmitted equally in all directions. Thus if small frictionless pistons working in cylinders be attached to the vessel in any position or direction, and each having the same area, say 1 square inch, then if any one of these be pushed inwards with a force of say 10 lbs., each of the others must have the same force of 10 lbs. applied to it to prevent it moving outwards.

If, now, two of these small pistons be connected or merged into one, consequently having an area of double the original or 2 inches instead of 1, the pressure required

is that of two of the original pistons or 20 lbs. In the same way, if two pistons be applied to the vessel, one having an area one hundred times that of the other, then the pressure required to prevent motion of the large piston will be one hundred times that of the small piston, and *vice versâ*. If motion is allowed to take place, the small piston will move through a distance one hundred times that of the large piston, or in other words the velocity of the small piston will be one hundred times that of the large piston; thus, what is gained in force is lost in velocity.

This principle was so well understood by Pascal at the time of his discovery (A.D. 1664) that we cannot improve upon his own clear wording. "If a vessel full of water, closed on all sides, has two openings, the one a hundred times as large as the other, and if each be supplied with a piston which fits exactly, a man pushing the small piston will exert a force which will equilibrate that of a hundred men pushing the piston which is a hundred times as large, and will overcome that of ninety-nine. And whatever may be the proportion of these openings, if the forces applied to the pistons are to each other as the openings, they will be in equilibrium. Whence it appears that a vessel full of water is a new principle of mechanics, and a new machine for the multiplication of force to any required degree, since one man will by this means be able to raise any given weight, It is, besides, worthy of admiration that in this new machine we find that constant rule which is met with in all the old ones, such as the lever, wheel and axle, screw, etc., which is that the distance is increased in proportion to the force; for it is evident that as one of these openings is a hundred times as large as the other, if the man who pushes the small piston drives it forward 1 inch, he will drive the large piston backward only one hundredth part of that length."

Principle of Surfaces of Equal Pressure.—Whereas the above principle is entirely independent of the action of

gravity, the one about to be discussed is a direct consequence of gravity. This principle states that in any horizontal layer of a liquid at rest the pressure is the same at all points, and the intensity of that pressure is directly proportional to the depth of immersion.

The demonstration of this principle is very easy, as we may imagine a small cube of some substance having the same weight as water immersed at any depth, then, if this cube is to remain stationary in a horizontal direction, the forces acting upon its opposite faces must be equal. The intensity of the pressure corresponding to any depth is best ascertained by direct experiment. Pascal performed this experiment with an apparatus known as Pascal's vases. He used glass vases having detachable bases formed of sheet metal, which were placed in contact with the smooth edges of the vases, thus forming a water-tight joint. The vase was fixed vertically in mid-air, and the base placed in position. A fine string attached to the centre of the base passed upwards and over a pulley, and had a weight attached to its other end. The base was thus pulled upwards with a known force. On carefully admitting water to the vase, the level of the water rose until its weight produced a sufficient downward pressure to overbalance the weight, and so allow the escape of the water through the bottom of the vase. By noting the height of the water at the time of overbalancing, it was found that the balance weight has the same weight as a column of water having a horizontal area equal to the opening in the bottom of the vase, and a height represented by the height to which the water rose in the vase.

Various shapes of vases were tried, some expanding from the base, and others contracting. The result was always the same, and was entirely independent of the total weight of water in the vase, but directly dependent upon the height to which the water rose. Thus we may have a hole of say 3 inches diameter, containing a diaphragm which is pressed upwards with a sufficient force to balance a column of water

30 feet high, and the pressure required is the same, whether the hole be in the bottom of a lake or a tube which contracts until its diameter is only 1 inch or less.

Fig. 1 illustrates this principle. Suppose the small plugs or pistons shown in the tube to be of negligible weight and frictionless, then the pressure in pounds to be exerted on each plug to prevent motion is found by measuring the area of

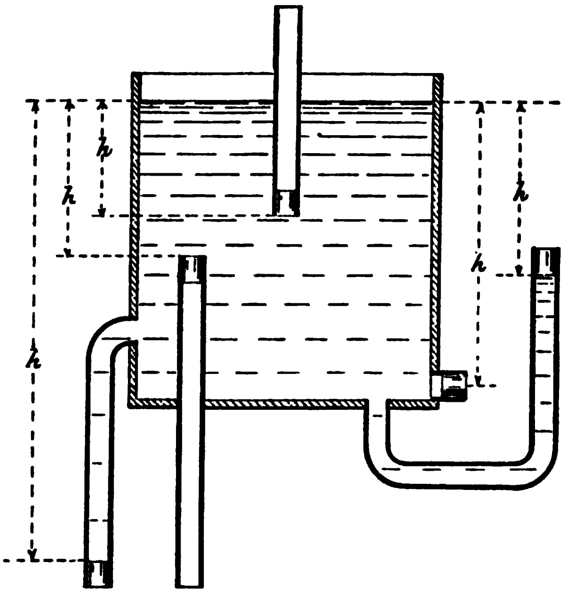


Fig. 1.

the plug in inches and multiplying by the corresponding height *h* in inches, as shown in the figure, and by the weight of 1 cubic inch of water. The tubes are all shown of parallel bore, but it matters nothing what shape of tube is used, nor how many contortions it makes before arriving at the plug.

Taking the weight of water as 62.5 lbs. per cubic foot, or

.434 lbs. per 12 cubic inches, we arrive at the following values, in which h represents the height or head in feet:—

Pressure per sq. foot - - - $= p = 62.5\,h$.
" " inch - - - $= p = .434\,h$.
Height due to pressure p per sq. foot $= h = .016\,p$.
" " per sq. inch $= h = 2.304\,p$.

By the above principle we are also enabled to ascertain the pressure acting against a vertical plane due to water at rest reaching any height up that plane or to some

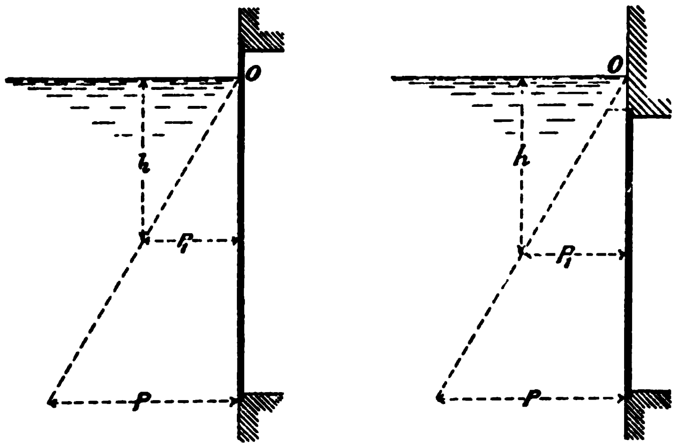


Fig. 2.

height above it. As the pressure is directly proportional in any horizontal plane to the height of water above that plane, we may calculate the pressure corresponding to the bottom edge of the vertical plane, and represent this pressure by the length of a line P drawn at right angles to the plane as shown in Fig. 2. By now drawing a sloping line joining the extremity of this line to the point O, where the surface of the water meets the vertical plane, and measuring the horizontal lengths joining the plane to the sloping line, we have the pressures correspond-

ing to any levels. By adding up these pressures ascertained for narrow horizontal strips the total pressure on the plane is obtained. This is the same as finding the immersed area of the plane, say in square feet, and multiplying by the pressure P_1 ascertained for 1 square foot at a depth corresponding to the depth of immersion of the centre of gravity of the immersed area of the plane—

$$\text{Area} \times P_1 = \text{total pressure.}$$

Principle of Archimedes.—About the year 250 B.C. Archi-

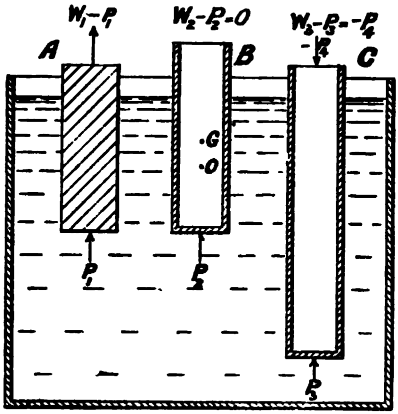


Fig. 3.

medes made the discovery that if bodies are immersed in water they lose in weight, and the amount of that loss is represented by the weight of the water displaced. Thus any body having 1 cubic foot capacity when immersed in distilled water loses weight to the extent of 62.25 lbs. When once the body has passed below the surface of the water, the depth to which it is afterwards immersed makes no difference to the truth of the principle, for though by the principle of surfaces of equal pressure there is an increasing upward pressure applied to the body by the water as its immersion

becomes greater, there is also a correspondingly increasing downward pressure.

Fig. 3 is a practical illustration of this principle in a form constantly met with in hydraulic machinery. Three bodies, A, B, C, are shown partly immersed in water. A is a solid cylinder of iron, having the weight W_1 when weighed in air. When immersed, as shown, there is an upward pressure P_1 due to the weight of water displaced, so that if a cord were attached to the iron cylinder A to prevent it sinking, the tension in the cord would be $W_1 - P_1$. The body B represents a hollow cylinder of iron which is immersed to such a depth that it floats. In this case the weight W_2 acting downwards is balanced by the pressure P_2, due to the water displaced acting upwards; consequently $W_2 - P_2 = O$. C represents a hollow iron cylinder immersed to a depth such that the upward pressure P_3, due to the water displaced, is greater than the weight W_3 of the cylinder. In this case $W_3 - P_3 = -P_4$, where P_4 represents the magnitude of a downward pressure necessary to prevent the cylinder rising to such height that $W_3 = P_3$ at which the cylinder would float as in the case of B.

A point worthy of consideration in connection with floating bodies is whether the body is in a state of stable or unstable equilibrium. In order to find whether the equilibrium is stable or otherwise, it is necessary to find the centre of gravity G of the floating body and the centre of buoyancy or centre of gravity O of the water displaced. If G is above O as shown in the figure the equilibrium is unstable, whereas if G is below O the equilibrium is stable. In the case shown at A the equilibrium is always stable, while in the case shown at C the equilibrium is always unstable.

The Barometric Column. — The phenomenon of the barometric column was first investigated by Galileo, who found that the greatest height to which water will stand in a tube from which the air had been exhausted is about 34

feet. Torricelli made further experiments and also used mercury. He pointed out that for a tube of any area the height to which a liquid stands is such that the weight of liquid column in the tube is always the same, no matter what liquid is employed, and that this weight represents the pressure of the atmosphere on the area of the tube. The average pressure of the atmosphere ascertained by this method is 14.7 lbs. per square inch.

The heights to which water will stand in a closed tube for various altitudes and atmospheric pressures are :—

34 feet	corresponding	to	14.7 lbs.	=	pressure at	sea-level.
31.7	,,	,,	13.7 ,,	=	,,	1,880 feet.
30.6	,,	,,	13.2 ,,	=	,,	2,870 ,,
29.5	,,	,,	12.7 ,,	=	,,	3,900 ,,

Theoretical Hydraulics.—The first point to be considered under this head is the principle of continuity of flow. If water is flowing through a pipe with any velocity, and the flow is to be continuous, the same quantity Q of water must pass any points we may choose in the tube in the same space of time. If v represents the velocity of flow, and A the cross sectional area of the tube, the quantity Q may be represented as $A \times v$, and this is true for all points in the tube. Hence whenever there is continuity of flow we have the equation—

$$Q = Av.$$

Instead of a tube of uniform cross section, a tube of varying cross section may be used, and consequently there will be a change of velocity. A diminution of area causes an increase of velocity and *vice versâ*.

$$Q = Av = A_1 v_1 = A_2 v_2, \text{ etc.}$$

Velocity due to Head.—The phenomenon of water flowing when subjected to a head or pressure has been made use of from the earliest times, but the law governing this velocity was investigated by Torricelli in A.D. 1644. Torricelli announced the law that, when water is subjected to a

head or pressure and allowed to flow unrestrained, the velocity of the water is the same that a body would acquire in falling through a height corresponding to the head of water producing the flow. If the velocity be represented by v feet per second and the height or head of water by h feet, then—

$$v = \sqrt{2gh}. \qquad g = 32.2.$$

In ascertaining the velocity of flow from an orifice in a vertical plane it is usual to take the height h as measured from the centre of gravity of the plane area of the opening. This method is not strictly correct, but for a head of three times the depth of the opening the error amounts to only 1 per cent., and for greater heads the error is less.

If the velocity is known and it is required to find the head producing the velocity, the above equation may be written—

$$h = \frac{v^2}{2g}$$

The head h is often referred to as the pressure head, and the quantity $\frac{v^2}{2g}$ as the velocity head.

Although the above equation is all that is required in reference to velocity of outflow from orifices, it does not state the conditions existing within the vessel containing the water. If the vessel is of larger cross sectional area than the orifice, then the velocity in it will be less than the velocity of outflow, while if at any part the vessel is contracted so as to have a cross sectional area less than the orifice, the velocity at that part becomes greater even than the velocity due to the head. This latter condition was observed by Bernoulli in A.D. 1738. Venturi made further experiments in A.D. 1791, and observed that an increase of velocity was accompanied by a decrease of pressure in the tube or vessel below the pressure of the atmosphere. There is of course a limit to this increase of velocity, that limit being reached

when the pressure in the tube becomes zero, or when a complete vacuum prevails.

Experiments conducted on tubes having a gradually changing cross sectional area show that where the tube is large, and the velocity of flow in consequence small, the pressure in the tube rises, until if the tube becomes so large that the velocity of flow is almost *nil*, the pressure approaches very nearly to that of the head producing the flow through the pipe. On the other hand, when the area of the tube contracts, the pressure falls. If these pressures and the corresponding velocities are noted, it is found that the amount by which the pressure falls below that due to the head is the amount of pressure head necessary to produce the velocity occurring in the tube. Written as an equation—

$$h - h_1 = \frac{v_1^2}{2g}, \text{ or } h = h_1 + \frac{v_1^2}{2g}$$

As this is true for any part of the tube, the equation may be written—

$$h = h_1 + \frac{v_1^2}{2g} = h_2 + \frac{v_2^2}{2g}, \text{ etc.,}$$

which is known as the hydrodynamic equation.

The Energy of Water.—There are three ways of expressing the energy of a quantity of water. In the first place, the water may be stored at a height above the level at which it is to be employed to perform work, the energy existing in the potential form. In the same way that, if a heavy body be sustained at some height, its potential energy may be expressed in foot-pounds by multiplying the weight of the body in pounds by the height in feet, so the potential energy of water may be expressed

$$Wh = \text{potential energy.}$$

Instead of the head being given, it is often stated that the water is at a certain pressure per square inch. In this case the energy per pound may be expressed by multiplying the

pressure per square inch by the length in feet of a column of water weighing 1 lb., and having a cross sectional area of 1 inch. Suppose a cylinder of 1 square inch area to contain a piston which is driven forward by water under a pressure p pounds, when the piston has moved forward 2.304 feet, 1 lb. of water has passed into the cylinder, and the work done is represented by $p \times 2.304$ foot-pounds. Thus the pressure energy of 1 lb. of water is $p \times 2.304$ foot-pounds or for any weight of water—

$$W \times p \times 2.304 = \text{pressure energy}.$$

It has already been pointed out that the height h due to a pressure p pounds per square inch is $2.304p$ feet. Therefore—

$$\begin{array}{ccc} Wh & = & W \times p \times 2.304. \\ \text{Potential} & = & \text{Pressure} \\ \text{energy.} & & \text{energy.} \end{array}$$

The third expression for the energy of water is used in the case of flowing water. It is well known in connection with falling bodies that the energy stored in the body in the kinetic form, due to the body having fallen freely from some known height, is ascertainable from the velocity acquired by the body in falling, and is represented by the equation—

$$W\frac{v^2}{2g} = \text{kinetic energy}.$$

It has already been stated that the velocity acquired by water under a head h is the same as that of a body falling freely through the distance h, hence the kinetic energy of a weight of water W is ascertained from its velocity by the above equation. By the principle of the conservation of energy, the potential energy must equal the kinetic energy, or—

$$Wh = W\frac{v^2}{2g},$$

which is easily proved since $v^2 = 2gh$, as already pointed out under *Velocity due to Head.*

If an inspection be now made of the hydrodynamic equation, we see that by multiplying each side by W the equation becomes—

$$Wh = Wh_1 + W\frac{v_1^2}{2g}$$

From the equation in this form it is noticed that the

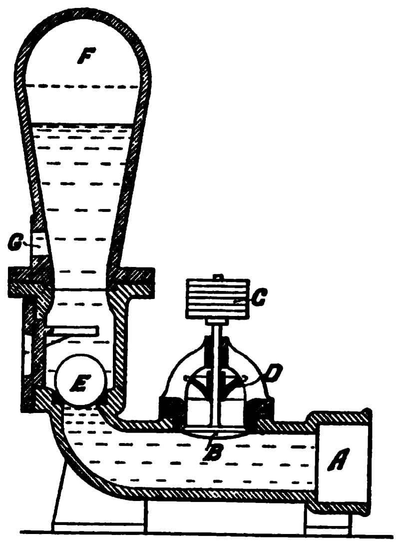


Fig. 4.

energy may occur partly as potential energy and partly as kinetic energy, or partly as pressure energy and partly as kinetic energy, as Wh_1 may be written $Wp_1 \times 2.304$. It is very important that this fact should be grasped at this stage, as there are very few hydraulic machines in which the energy does not occur in this form while the machine is at work.

The relation existing between the different forms in which

the energy may occur can be rendered more clear by an examination of the working of the hydraulic pump, commonly known as the hydraulic ram, illustrated in Fig. 4. The object of the apparatus is to pump water to a considerable height by utilising the potential energy of a supply of water placed at a smaller height. At the joint A connection is made to a length of pipe, usually 10 to 20 feet, leading to the supply of water to be utilised. Connection is made at G to the receiving tank to which the water is to be pumped, so that the air contained in the bell F is compressed to a pressure corresponding to the head of water connected to G. When the valve B is shut the water in the pipe A is stationary. The weights C applied to the valve B are sufficient to overcome the pressure in the pipe A and thus cause the opening of the valve. The water now begins to acquire a velocity and escape through the valve B, thus converting the whole or part of its potential energy into kinetic energy. As the water escapes through the valve B it meets the guide D and is deflected, causing an upward pressure on the valve spindle sufficient to overcome the weights C and close the valve. The water in the pipe A, having a velocity and corresponding kinetic energy, is now entrapped in the pipe, and as this energy cannot be dissipated and cannot continue wholly in its present form, as the velocity of the water has been checked, it is evident there must be a conversion of energy to the pressure form.

This conversion causes a heavy pressure to be generated in the pipe A, and when this pressure has risen above the pressure in the chamber F the ball valve E will be raised, and water will flow from A to F as long as the pressure is maintained in the pipe A greater than the pressure in F. During the entry of the water from A to F the pressure in F is increased owing to the compression of the air. This increase of pressure overcomes the pressure acting at G, and there is a consequent flow through G to the elevated

tank. On the closing of the valve E the pressure in A again returns to that due to the smaller head, and the valve B is free to be operated by the weights C causing a repetition of the operation. Thus we have converted potential energy to kinetic, kinetic to pressure, and pressure to potential energy.

The Reaction of Flowing Water.—When water is flowing from an orifice with a velocity due to some head of water, we have noticed that the velocity v is the same that would be acquired if each particle started from the upper surface of the water and fell freely under the influence of gravity. It is possible to calculate the magnitude of a force F which, acting for one second on the weight W of water flowing per second, would cause it to acquire the velocity v. As F acts for one second the distance through which it acts is $\frac{1}{2}v$, and the equation may be written—

$$F\frac{v}{2} = W\frac{v^2}{2g}$$

$$F = \frac{Wv}{g}.$$

The expression $W\frac{v}{g}$ will be at once recognised as the usual formula for momentum. As W may be written wav, in which w is the unit weight of water and a the area of the orifice, the formula becomes—

$$F = \frac{wav^2}{g} = 2wa.\frac{v^2}{2g}$$

in which $\frac{v^2}{2g}$ may be substituted by h, so that—

$$F = 2wah.$$

As wah represents the weight of the column of water producing the velocity, the force F is equal to twice the weight of the column.

Several experiments have been performed to demonstrate

the above fact. In one form the jet of water is allowed to meet a plane, when the plane is urged away from the jet with the force F as above calculated. In another form the plane is placed against another orifice subjected to a head of twice that producing the jet, when it is seen that the jet retains the plane in position, thus keeping back the greater pressure by the reaction force F.

CHAPTER II.

THE OBSERVED FLOW OF WATER.

THE remarks upon the flow of water in the last chapter had reference to the theoretical velocities, and no allowance was made for loss by friction and other causes. These losses must now be investigated before the formulæ there given can be successfully applied to the design of hydraulic machinery.

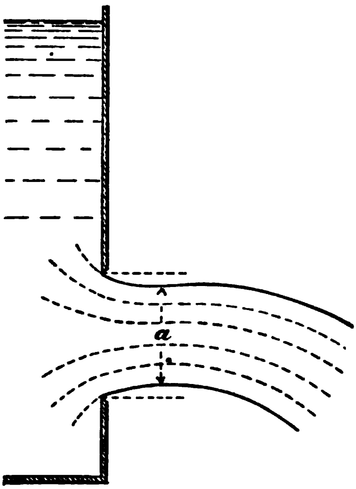


Fig. 5.

The attempt to ascertain the exact quantity of water flowing through an orifice has been the cause of a large number of experiments being performed. Fig. 5 shows the orifice as usually arranged, the edges being chamfered off so as to produce a sharp line in contact with the water. The orifice may be cut in a piece of hardwood or in thin metal. As

these orifices are largely employed in accurately measuring the water flowing from a hydraulic machine under trial, and for other similar purposes, it is essential that some standard should be fixed in order that the exact quantity of water flowing per second may be computed from tables compiled from well-authenticated experiments. It is found that if the inner edge of the orifice is rounded off, the flow is subject to alteration for a comparatively small difference of form, hence the sharp edge is always employed.

In using an orifice the vessel should be considerably larger than the orifice, in order that the velocity of approach may be small compared to the velocity of discharge. For the same reason the head should not be too small. As the water issues from the orifice a contraction takes place, known as the contracted vein, so that the effective area of the orifice is less than the measured area. The values of the coefficient of contraction have been assigned by different authorities as ranging between .71 and .60, generally .63, of the measured area.

The velocity of flow at the contracted area *a*, Fig. 5, should be the velocity due to the head, but owing to frictional losses it falls to values of .99 to .97 of the theoretic value. These values are called the coefficients of velocity.

The most important point to settle is the coefficient of discharge; the quantity of water actually flowing can then be ascertained by multiplying the quantity due to the area of the orifice and the theoretic velocity by this coefficient—

$$cQ = q.$$

VALUES OF *c* (FROM HAMILTON SMITH'S TABLES).
Circular Orifices (Vertical).

Head in Feet.	Diameter of Orifice in Feet.						
	.02	.04	.07	.10	.20	.60	1.0
1	.644	.623	.612	.608	.600	.595	.591
10	.611	.603	.599	.598	.597	.596	.595
100	.593	.592	.592	.592	.592	.592	.592

Square Orifices (Vertical).

Head in Feet.	Side of Square in Feet.						
	.02	.04	.07	.10	.20	.60	1.0
1	.648	.628	.618	.613	.605	.601	.599
10	.616	.608	.605	.604	.603	.602	.601
100	.599	.598	.598	.598	.598	.598	.598

Rectangular Orifices, 1 foot wide (Vertical).

Head in Feet.	Depth of Orifice in Feet.						
	.125	.25	.50	.75	1.0	1.5	2.0
1	.632	.632	.618	.612	.606	.626	...
10	.606	.603	.601	.601	.601	.601	.602

It must not be supposed, because there is a great difference between the discharge from an orifice and that calculated from the area of the orifice, that there is a corresponding loss of energy. The loss of energy is given by the coefficient of velocity, and as the energy is proportional to the square of the velocity, assuming the coefficient of velocity to be .98, the energy is $\frac{1}{2g}(.98v)^2 = .96v^2\frac{1}{2g}$. This is an efficiency of 96 per cent., or a loss of 4 per cent.

The quantity of water flowing may also be measured by means of a weir. Fig. 6 shows a weir fixed in a stream for the purpose of measuring the supply of water. There are two kinds of weir usually employed. One consists of a rectangular notch considerably narrower than the stream, so that the water may approach it freely in all directions. In the second form the weir is suppressed at the ends by boards

such as A, so that the water flows in parallel lines so far as the lateral directions are concerned, but free approach is permitted from below. In taking the height of water above the weir the operation should be conducted some distance back from the weir, as the upper surface of the water slopes in a direction towards the weir.

The method of gauging with a rod, as shown, is only suited for large heads and for rough estimates. The best method

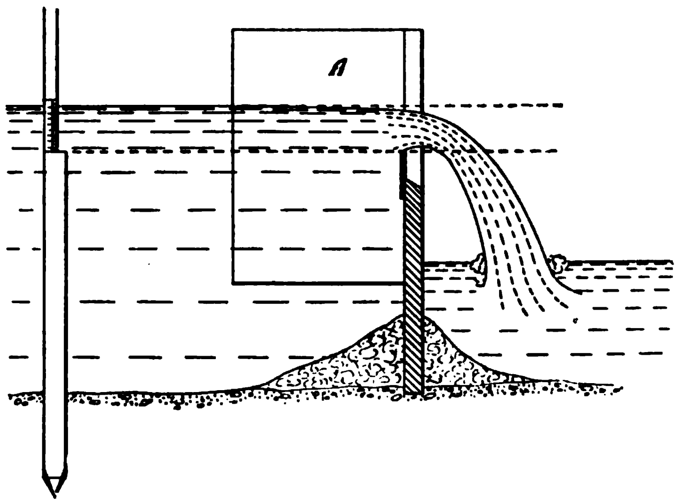


Fig. 6.

of measuring the head is with the hook gauge, invented by Boyden in A.D. 1840, which consists of a rod having a scale marked accurately and reading by the aid of a vernier to ten-thousandths of a foot. The bottom end of the rod is fitted with an upturned point, which is adjusted to the level of the water when the bottom edge of the weir has been reached but no flow is taking place. The vernier is now set to zero, and when the water has reached the maximum height above the weir the rod is carefully raised by means of a worm-wheel

and thumbscrew until the point just touches the surface. A second reading is now taken, and the height of the water is at once ascertained. The point can be accurately set to the level of the water, as if lifted too high a pimple is formed on the surface of the water due to capillary attraction.

The flow taking place over weirs may be calculated from the equation—

$$q = c.\frac{2}{3}\sqrt{2g}.bH^{\frac{3}{2}},$$

in which b represents the length of the weir in feet, and H the height measured by the hook gauge. Numerous experiments have been performed to ascertain the value of the coefficient of discharge c.

VALUES OF c (FROM HAMILTON SMITH'S TABLES).

Head in Feet.	Length b in Feet.						
	.66	1.0	2.0	3.0	5.0	10	19
.1	.632	.639	.646	.652	.653	.655	.656
.2	.611	.618	.626	.630	.631	.633	.634
.3	.601	.608	.616	.619	.621	.624	.625
.4	.595	.601	.609	.613	.615	.618	.620
.6	.587	.593	.601	.605	.608	.613	.615
.8	...	...	.595	.600	.604	.611	.613
1.0	...	...	.590	.595	.601	.608	.611
1.4	...	...	.580	.587	.594	.602	.609

If there is a noticeable velocity of approach where the hook gauge is placed, the above formula must be modified as follows:—

$$q = c.\frac{2}{3}\sqrt{2g}.b(H + 1.4h)^{\frac{3}{2}},$$

in which h represents the head producing the observed velocity of approach, c having the same values as before. In selecting c the new head $H + 1.4h$ must be used.

When the weir is suppressed by the boards A the same equation applies as for the free weir if there is no velocity of approach, but different values of c must be used—

$$q = c.\frac{2}{3}\sqrt{2g}.b\mathrm{H}^{\frac{3}{2}}$$

VALUES OF c (FROM HAMILTON SMITH'S TABLES).

Head in Feet.	Length b in Feet.						
	2	3	4	5	7	10	19
.1	...	...	...	.659	.658	.658	.657
.2	.645	.642	.641	.638	.637	.637	.635
.3	.639	.636	.633	.631	.629	.628	.626
.4	.636	.633	.630	.628	.625	.623	.621
.6	.638	.634	.630	.627	.623	.620	.618
.8	.643	.637	.633	.629	.625	.621	.618
1.0	.648	.641	.637	.633	.628	.624	.619
1.4	...	...	.644	.640	.634	.629	.622

When there is a perceptible velocity of approach the equation becomes—

$$q = c.\frac{2}{3}\sqrt{2g}.b(\mathrm{H} + 1.33h)^{\frac{3}{2}},$$

and c must be found from the table corresponding to the head $\mathrm{H} + 1.33h$.

The flow from a short tube, usually about three diameters in length, called the standard tube, is very instructive, and is of practical interest to the hydraulic engineer. This tube, Fig. 7, is arranged in the side or bottom of a vessel, and has a perfectly sharp inner edge, as in the case of the orifice. It is found by experiment that the discharge from a short tube is greater than from the orifice of similar diameter, but the velocity of outflow is considerably less. As the water completely fills the tube where it is discharged, the coefficients of

velocity and discharge are always equal. The values of these coefficients vary from .83 to .80, decreasing for larger heads.

If the tube be made of glass or other transparent material, it is noticed that there is a contracted vein occurring within the tube. The tube is very inefficient owing to the low value of the coefficient of velocity, and the energy of the issuing stream may be found as in the case of the orifice, and

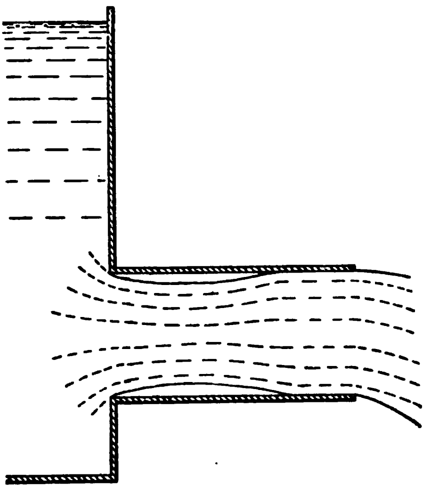


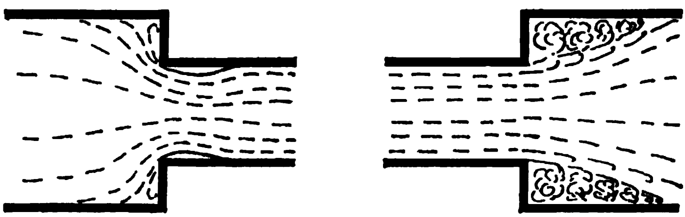
Fig. 7.

is represented by $\frac{1}{2g}(.82v)^2 = (.67v)^2\frac{1}{2g}$. This is an efficiency of 67 per cent., or a loss of 33 per cent.

The low efficiency of the standard tube has caused experiments to be made with coned tubes, with the result that much higher velocities have been obtained. The cone is described by the angle which one of its sides produced would make with the centre line; thus a cone of 10° angle has a total convergence of 20°. Experiments on these cones show

that as the angle is increased from 0° the coefficient of discharge increases from about .82 to .946, corresponding to an angle of 13°-24′, when it again decreases. The coefficient of velocity, however, continues to increase until for an angle of 48°-50′ it has a value of .984. This continued increase appears to suggest that the highest coefficient is obtained with a cone having the form of the contracted vein due to the velocity corresponding to the head of water available.

There are two arrangements of supply pipes which concern the hydraulic power engineer; first, where it is desired to conduct the water from an elevated reservoir to work a turbine, and secondly, where high-pressure water is discharged into a pipe to be consumed in working hydraulic



Figs. 8 and 9.

machines placed in various positions. In the first case the same quantity of water flows from end to end of the pipe, whereas in the second, the quantity is reduced by being abstracted by branch pipes leading to the machines. In all cases it is essential that there should be as small loss as commercial circumstances will permit. The losses occurring are caused by eddy currents due to the sudden change of section of the pipe, or to bends, and by friction of the water against the sides of the pipe.

The losses due to change of section may be explained with reference to Figs. 8 and 9. If the section is reduced suddenly the conditions of the standard tube obtain with the consequent loss of efficiency. This evil may be largely

remedied by substituting a cone and keeping the velocity in the pipe low. When a sudden enlargement occurs the loss is caused by the water whirling round, and if v be the greater velocity, and v_1 the reduced velocity, the loss of head may amount to—

$$h_1 = \frac{1}{2g}(v^2 - v_1^2).$$

This loss may, however, be largely prevented by the use of a cone.

The friction of water in a pipe is found to vary directly as the square of velocity of flow, and the length of the pipe, and inversely as the diameter of the pipe, also directly as a coefficient which is reduced for an increase of velocity—

$$h_1 = f_o . \frac{l}{d} . \frac{v^2}{2g},$$

all the dimensions being in feet.

VALUES OF f_o FOR SMOOTH IRON PIPES.

Diameter d.	Velocity v.					
Feet.	1.0	2.0	3.0	4.0	6.0	10.0
.05	.047	.041	.037	.034	.031	.029
.1	.038	.032	.030	.028	.026	.024
.25	.032	.028	.026	.025	.024	.022
.5	.028	.026	.025	.023	.022	.020
.75	.026	.025	.024	.022	.021	.019
1.0	.025	.024	.023	.022	.020	.018
1.5	.023	.022	.021	.020	.018	.016
2.0	.021	.020	.019	.017	.016	.014
3.0	.019	.018	.016	.015	.014	.013
4.0	.017	.016	.015	.013	.012	.011
6.0	.015	.014	.013	.012	.011	...

When the theoretic head in a parallel pipe has been diminished in accordance with the above formula, instead of having the same value for any part of the pipe, it is found

to decrease in the direction of flow; this decrease is known as the hydraulic gradient.

In the previous chapter it was pointed out that the reaction force of flowing water is equal to the weight of twice the column producing the velocity of flow, and as this fact has

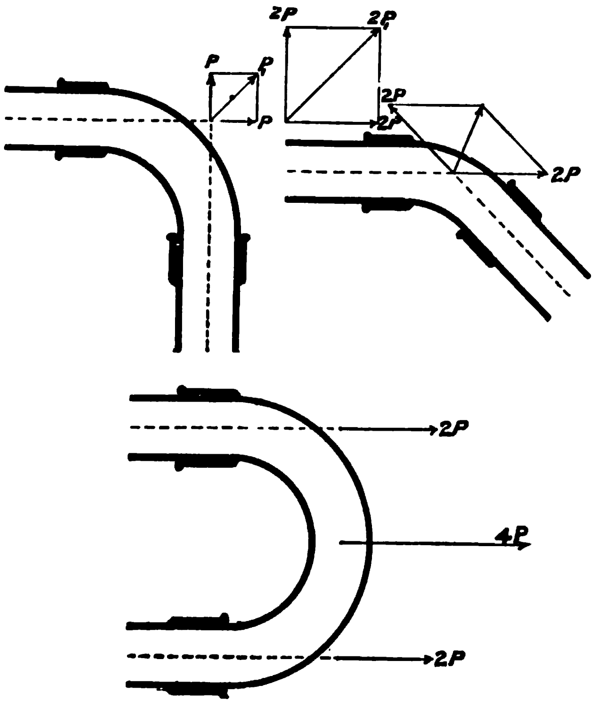


Fig. 10.

an important bearing on several branches of hydraulic design, it is worthy of further consideration. As an example, we may consider a pipe having a right angle bend in which the water is stationary, and the total pressure due the head and pipe area = P. By referring to Fig. 10 we see that

there are two pressures P tending to force the pipes to part at the joints. If the water be now allowed to flow with the velocity due to the head producing P, these pressures P are at once changed to 2P by the reaction. To prevent the joints parting, a concrete block or other obstruction must be built in contact with the bend, and the force against this obstruction is the resultant $2P_1$ of the forces 2P.

The magnitude of the force $2P_1$ may be found in the same way for any other bend in the pipe, either greater or less than a right angle. When the bend is 180°, the force $2P_1$ becomes 4P.

If the velocity of flow does not represent the total head, the force 2P becomes—

$$P_0 + 2P,$$

in which P_0 is the total pressure due to the pressure head and pipe area, while P is the total pressure due to the velocity head and pipe area. The meaning of pressure head and velocity head have already been given.

Instead of a pipe bend the water may be caused to flow against a curved vane or guide, when the pressures are identical with those above considered.

PART II.—PRELIMINARY.

CHAPTER III.

HYDRAULIC PRESSURES.

BEFORE proceeding with an examination of the principles connected with hydraulic power in its application to machinery, it is desirable that the more general principles which govern the employment of the various members or parts when placed in combination in any one machine shall be understood, and the fitness of the respective parts for the duties required inquired into.

In the description and illustration of what we may term the elements—that is, the component parts or details of machines—we shall be led to introduce much information, which to the experienced hydraulic engineer or draughtsman will no doubt appear superfluous. The more experienced reader should, however, bear in mind that to many practical engineers the conditions and mode of working, the details of construction, the soundness or unsoundness of various arrangements, and even the general principles of action of hydraulic machinery, are a true *terra incognita*, while to the younger engineers and draughtsmen the more fundamental portion of our description may not be the least valuable.

We shall, then, first take up the consideration of the elements, the details of construction, of hydraulic machinery, commencing with the simplest parts, such as the valves and seatings, various types of packings (their friction and best method of construction), pipes, joints, glands, safety valves, stop valves—passing from the simplest screw-down valve to the more complicated types which command the whole action of a complex machine by the movement of a single lever, and may almost be considered machines in them-

selves—and the various other details, the correct design and construction of which are of importance as affecting the permanence, economy, or safety of the machine.

For the proper consideration of the subject, it is absolutely necessary to divide hydraulic machinery operated by pressure energy into at least three classes, defined according to the intensity of the pressure by which they are operated. This is due to the extended range of pressure adopted in the working of different types of hydraulic machines. Thus, the author has designed hoists which work successfully and with fair economy with a water pressure of only 5 lbs. per square inch, and on the other hand plants for testing the internal steel tubes of modern ordnance to the intense pressure of 1,000 kilogrammes per square centimetre, or about 6½ tons per square inch, a pressure equivalent to that of a column of water nearly 6½ miles high.

The contrast between steam and hydraulic machinery is in this respect very striking. Whereas in the case of hydraulic work we have a range of pressure of from say 5 lbs. to 22,400 lbs. per square inch, necessitating considerable modification in the details of construction and choice of material for the various parts, we have in the case of steam machinery a maximum practical range of pressure of from 7 lbs. per square inch to 300 lbs. per square inch only, and the small modification of construction and material of detail at the higher pressures is due more to the difference of temperature of the steam than to difference in its pressure; a difference of temperature which, on the other hand, does not occur in the case of hydraulic machinery.

We shall, then, divide hydraulic pressure machinery into three classes:—

1. *Low Pressure.*—Comprising all machines intended to work with a pressure of less than 200 lbs. per square inch.

2. *Medium Pressure.*—Comprising machines intended to work at a pressure of from 400 lbs. per square inch to 1,500 lbs. per square inch.

3. *High Pressure.* — Comprising machines intended to work at pressures of from 1 ton to 10 tons per square inch.

The low-pressure class is largely used in the operation of hydraulic lifts for hotels, etc., steam or gas engines being fixed in the basement to supply the requisite pressure, either direct or by pumping into a tank on the roof of the building, at a sufficient height to furnish an adequate head for working the lift. Hoists supplied with pressure from the water supply mains of the town also fall into this class, and are largely used. It is an excellent practice, in the case of large works and manufactories in which fire-mains are laid down and steam pumps fixed to supply them, to keep the pumps running as required throughout the day instead of standing idle, and utilise the pressure in the operation of hoists of this first class throughout the establishment.

The medium-pressure class includes the Armstrong type. The pressure originally employed by Mr Armstrong (now Baron), who may be considered the foster-parent of the system of working an entire plant of lifting and hauling machinery by hydraulic pressure generated at some convenient centre and distributed by mains, being 700 lbs. per square inch—a very suitable pressure for dock and station work and many descriptions of hydraulic machines, and adopted as a mean pressure by the hydraulic-power companies of London, Manchester, and Hull for their extensive plants for the distribution of power to consumers by mains laid beneath the public streets. 700 lbs. per square inch is, however, objectionably low, and even absolutely inadmissible for direct use in powerful hydraulic presses—generally involving the additional complication of an intensifier—and too high for simple application to direct-acting lifts when the height of lift is considerable. Medium pressures of from 700 lbs. to 1,500 lbs. per square inch are usually employed in connection with hydraulic riveting plants of the Tweddell and other types.

With reference to the third class, working at pressures of from 1 ton to 10 tons per square inch, a higher pressure than 2 tons to 3 tons per square inch is not to be recommended for permanent machinery and plants. For small apparatus, such as punching bears for boiler and ship work, where portability is one of the most sought for qualities of the machine, a pressure of 4 tons per square inch may be adopted with fair success, but pressures exceeding 3 tons per square inch should never be employed unless the conditions of the case are in great measure compulsory.

The reasons why pressures of say from 3 tons per square nch to 7 tons per square inch cannot be used with such practical success as lower ones do not arise so much from any difficulty in making joints, valves, or rams initially free from leakage, or in obtaining sufficient strength in the cylinders to resist the intense pressure; but lie in the rapid wear of the ram packings, from causes which will become apparent when we consider the action of the various packings available, as we propose to do in a subsequent chapter; and also in the rapid deterioration of the valves and valve seats. At these high pressures, when once a current (however infinitesimal) is established past the valve seat, either through the lodging of a minute particle of some hard substance on the seat or other cause, the water cuts rapidly into the metal of the valve or valve seat, sometimes forming a straight groove, sometimes a curious crooked one. In a very short time a large plant may be rendered useless for a time from this cause.

To comprehend the nature of this erosion of the hardest metals by a current of water, it is necessary to consider the enormous velocity at which water will pass through an aperture at these high pressures. The table below gives the velocity in feet per second, at pressures of from 700 lbs. to 10 tons per square inch, which the molecules of water will acquire if discharged into a vacuum through an approximately frictionless aperture. Velocities

for intermediate pressures may be calculated from the formulæ—

Velocity in feet per second = $12.19\sqrt{\text{pressure in lbs. per sq. in.}}$,
and velocity in feet ,, = $577\sqrt{\text{pressure in tons per sq. in.}}$

Velocity in feet per second...	326	472	577	707	816	912	999	1290	1527	1824
	lbs.		Tons.							
Pressure per sq. in.	700	1500	1	$1\frac{1}{2}$	2	$2\frac{1}{2}$	3	5	7	10

Thus the velocity of the molecules of water and any small particles of solid matter they may carry with them is as high at a pressure of 10 tons per square inch as the muzzle velocity of a modern gunshot, and the effect of this bombardment of the valves and seats may be compared to the action of the well-known sand blast. Even at a pressure of only 2 tons per square inch the velocity will be seen to be 816 feet per second, and will rapidly cause erosion or cutting if once a current is established. It is on this account advisable with all high-pressure hydraulic work to pass the water through a rough filter before reaching the pumps, in order that it may be as free from all solid particles as possible.

A pressure as high as 2 tons per square inch, however, may be successfully employed throughout a large establishment, and is indeed a very suitable pressure to adopt where capstan or other rotary engines are not needed, and the work required is mainly press work. We remember a well-known engineer asserting that it was impossible to work a large plant of hydraulic machinery with success at a pressure of 4,000 lbs. per square inch. This is, however, quite a mistake. The writer had under his personal observation for many years a large plant working at this pressure, and comprising forging presses, hoists, punching and shearing machines, boiler and girder riveters, and, in addition, the crane and hoist of a steelworks, and with perfect success.

It must, however, be admitted that to prevent failure careful attention is needed. Accumulator packings must be replaced at regular intervals, whether worn out or not, and all valves and seatings similarly examined at stated periods, and trued up or replaced if showing the least tendency to deterioration. If in addition ample pumping power and accumulators in duplicate be provided, a pressure of 2 tons per square inch can be adopted as confidently as a pressure of 700 lbs. per square inch, and has the advantage noted above of being much more suitable for heavy press work.

Three tons per square inch may be considered as the standard pressure adopted in the Manchester packing houses; many oil presses, and a considerable number of the large cotton-baling presses used in India, are also worked up to this pressure. In the case of such presses, however, an accumulator is rarely used for the maximum pressure, and pumps, valves, pipes, etc., are subjected to the extreme pressure only at the termination of the stroke of a press.

In treating of the materials used in the construction of hydraulic work, we shall make our remarks very brief, limiting them to such features as are of special importance in connection with hydraulic machinery. The general properties of such materials will be found so fully detailed in many works already in the hands of engineers, that it would be superfluous to recapitulate them here.

Cast iron is the metal most largely employed by the hydraulic engineer. It has, however, a reputation for want of reliability, especially in the construction of cylinders for high pressures. This reputation has been too often earned, however, by failure from improper disposition of the material, inadequate dimensions, or improper treatment in the foundry. As instances of the former fault, we illustrate below two typical cases, examples of which may frequently be met with in practice, even in the work of reputable engineers, and such as we have known to result in failure in more than one instance.

Figs. 11 and 12 represent a cross section through a cylinder (diameter = D), having a passage (diameter = d) cast on at

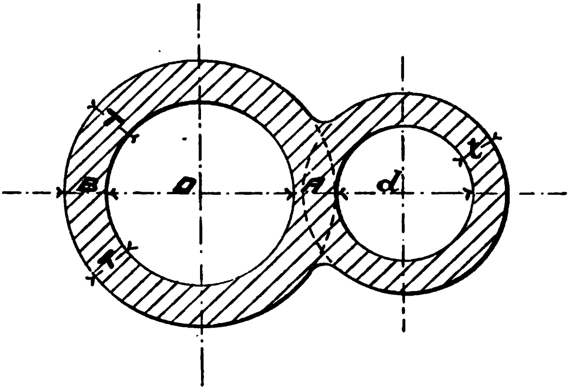


Fig. 11.

the side. Fig. 11 shows the faulty construction, and Fig. 12 the correct construction. In Fig. 11 it will be seen the designer has determined the thickness T of the cylinder D

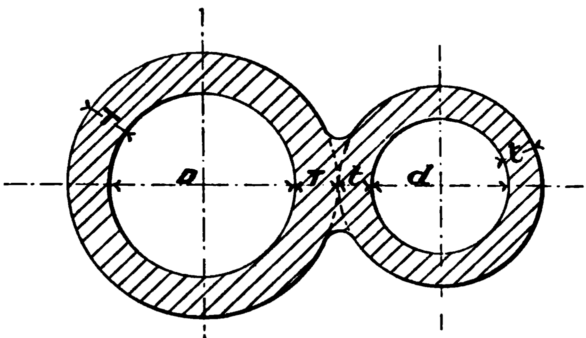


Fig. 12.

in the ordinary way, and then clapped on the passage d without considering the effect of the addition on the stress

on the metal between the cylinder and passage—that is at A. Thus, if P be the water pressure, the stress on the metal at A is $\frac{D+d}{2T} \times P$, while at B it is only $\frac{D}{2T} \times P$. Hence if the metal at B be properly proportioned to withstand the pressure P, the metal at A is decidedly too weak, and its thickness should have been $T+t$, as indicated in Fig. 12.

Fig. 13 similarly illustrates a faulty and a correct method of making the inlet-pipe connection to the side of a high-

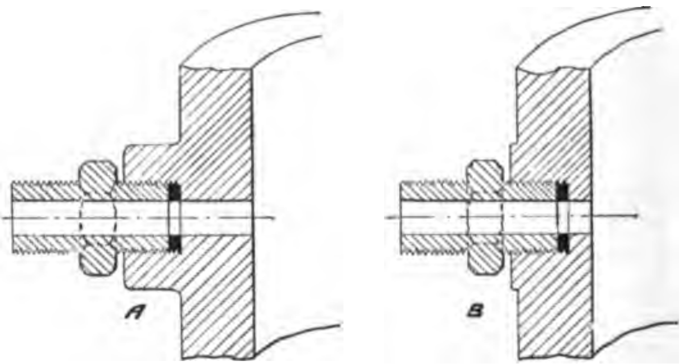


Fig. 13.

pressure cylinder. A is, of course, the correct construction, and B the faulty one. At B a large hole for the reception of the inlet nipple has been drilled and tapped, and only reinforced by a shallow boss, and, in some cases which have come under our notice, by no boss at all. At A only the comparatively small and necessary inlet hole penetrates the barrel of the cylinder, and the strength of the metal thus taken away is amply supplied by the substantial boss into which the inlet nipple is screwed. Such faulty constructions as those illustrated by Figs. 11 and 13 may stand the test pressure, and work without failure for a considerable time,

or, indeed, if there be ample material, may outlive the machine. On the other hand, if the thickness of the metal be originally somewhat inadequate, or the machine over-stressed through some accidental cause, weak points have been provided by the designer at which fracture may commence, causing, possibly, great loss and annoyance, and resulting simply from the want of a few pounds of metal *in the right place.* The construction illustrated at B is especially faulty owing to the intense stresses liable to occur at the edges of the nipple hole, owing to the break of continuity of the metal and consequent localisation of strain.

If, however, cast-iron cylinders be well and properly designed, cast from a suitable blending of metal, and with proper care on the part of the founder, they may be used with confidence for pressures up to 2 tons per square inch; and for thoroughly steady loads, such as those obtaining in the case of ordinary presses used in the compression of yielding and elastic substances, a pressure of 3 tons per square inch is not inadmissible.

CHAPTER IV.

MATERIALS.

THERE is, in general, no true economy in the employment of inferior metal in the construction of parts of machines in which great strength is required, since the loss of strength due to the inferior quality of the metal is far from compensated for by a slightly diminished first cost of the machine. In low-pressure hydraulic machines the thickness of the castings is frequently dictated by the exigencies of manufacture, and not by the working stresses to which they are subjected; but in the case of cylinders of medium pressure, and still more so in the case of the cylinders of high-pressure machines, which are frequently worked up to their full test pressure, or say one-half their probable initial breaking load, metal of first-class quality should invariably be employed. The cast iron for such purposes should be of at least such quality that a test bar 1 inch square, cast on end, will not break with a tensile load of 9 tons, and a bar 1 inch by 2 inches, placed on edge and carried by supports 3 feet apart, should sustain 30 cwt. in the centre without fracture. The metal, when cast into the actual shapes in which it is used, will in general have a considerably lower resistance to fracture than that of the test specimens, and it will not be wise to exceed a test stress on the metal of the complete machine of say $3\frac{1}{2}$ tons per square inch.

With respect to the working stress and factor of safety, as it is commonly called, we shall have something to say further on, as also as to the peculiar and dubious character of the stress sustained by thick cylinders under internal fluid pressure.

With regard to wrought iron, there is little to be remarked having special reference to hydraulic work. When used for cylinders, it must of course be thoroughly sound, and should not be designed for a higher test stress than 8 tons per square inch distributed, and if the thickness of the metal be considerable a lower stress may be advisable; a point we intend to discuss further on. For rolled Staffordshire bars of fair quality, a test stress of 10 tons per square inch is not too high, if applied in simple direct tension.

Steel is a material which has only lately come into general use for hydraulic cylinders, but the success which has rewarded the efforts of the steel-founder in the production of thoroughly sound and reliable steel castings is causing steel to rapidly replace cast iron in the construction of cylinders for high pressures. The breaking strength in tension of the metal employed is usually stated at 24 tons per square inch, but this is not probably obtained in the actual cylinder casting, the test stress on which it will be well to limit to 8 tons per square nch for cylinders of moderate thickness. For sound hammered steel cylinders, or hydraulic forged, a test of 10 tons per square inch of metal will not be too high.

Solid drawn steel tubes forms an excellent, indeed the best material available for high-pressure hydraulic tubes.

For the rams of hydraulic presses and hoists, rolled or hammered steel is frequently used, and sometimes steel castings, but there is a difficulty in getting the latter sufficiently sound on the surface for use in high-pressure work. Indeed, even in the case of hammered steel it is necessary to allow ample metal in the forging to permit of a substantial first cut being taken off over the surface (the rough should be at least $\frac{3}{8}$ inch larger in diameter than the finished ram), as otherwise it is impossible to eradicate the unsoundness due to the surface blowholes invariably found in the ingot. These, although closed in by the subsequent hammering, which leaves an apparently sound face in the finished use, are not really welded up, but reappear in the

shape of an unsound surface on the first cut being taken off in the lathe.

Malleable cast iron, toughened sometimes by the addition of a little scrap steel, is used with success for small short cylinders. Its ultimate strength, 1 inch thick, does not exceed in general 15 tons per square inch, and for ½ inch thick about 20 tons per square inch. The test stress may be taken at 8 tons per square inch, if the metal does not exceed ¾ inch thick. It is, however, a treacherous material, very liable to unsoundness, and should only be used for small and unimportant work.

The alloys of copper, tin, and spelter are of the greatest importance to the hydraulic engineer, owing to their freedom from corrosion by water. Hence they are used almost to the exclusion of any other metal for barrel linings, plungers, valves and valve seats, screwed caps and plugs, etc. Brass also forms an excellent sheathing for the outside of rams, and its use for that purpose is highly conducive to the durability of leather packings, while in all cases in which a cylinder is bored to receive a leather-packed piston it should also be lined with brass or gun metal, unless there be special circumstances which militate against their use. For the smaller class of pumps gun-metal castings are almost exclusively employed. The castings so used are in general somewhat, but not greatly, tougher and stronger than good cast iron. A test stress of 4 tons per square inch of metal may be permitted for gun-metal pump barrels.

For hydraulic pressures exceeding 4 tons per square inch steel should be used in place of gun metal. The portion of the brass foundry occupied in the production of hydraulic castings should be separate from that in which the commoner descriptions of metal are cast. Very annoying inequalities in the strength and closeness of the metal, due either to carelessness or wilful neglect on the part of the workmen, are otherwise extremely liable to occur. For pump plungers the rolled alloys, such as Kingston metal

and rolled phosphor bronze, are very reliable. These and similar alloys, in the form of rolled rods and solid drawn tubes, can now be procured of the strength of steel, and at very moderate prices.

Phosphor and manganese bronze castings are also used for pump barrels, and are said to have an ultimate breaking weight of about 19 tons per square inch of metal, but as far as the author's experience extends this cannot be depended on in the actual castings. The test stress for phosphor bronze pump castings may be taken at 6 to 7 tons per square inch of metal. Rams are coated with copper by electro-deposition by the Broughton Copper Company, of Manchester, and other firms, at very moderate cost. The finished thickness of copper usually supplied is $\frac{1}{32}$ inch. The durability of the sheeting so formed can be relied on, and its great gain in the first cost, as compared with brass sheathing, has brought this plan into favour.

Leather and one or two other materials of special utility for hydraulic work will be dealt with in connection with their applications.

Having considered the safe test stresses of the materials employed in hydraulic work, we have now to consider the not less important question as to what proportion the actual working stress should bear to that stress.

Very hazy notions on this subject have been held up to recent times, and, indeed, are still held. Great importance used to be attached to the determinations of the so-called "elastic limit" of a material, by which term was intended that stress at which the metal began to take noticeable permanent set. It was demonstrated by Mr Hodgkinson, however, that cast iron had no definite "elastic limit." By experiments with long cast-iron bars (15 feet long) he showed that cast iron takes a permanent set with small loads, increasing gradually, as the load is increased, up to the breaking point. Ductile wrought iron and mild steel have, however, a definite "elastic limit" of stress, or rather

they have a definite "breaking-down" point. This will be better understood by reference to the annexed diagram, Fig. 14, which represents the extension of a mild steel bar, 1 inch square, 10 inches long, under loads progressing in strain up to the breaking point. The author has carried out a very large number of experiments with mild steel bars, and

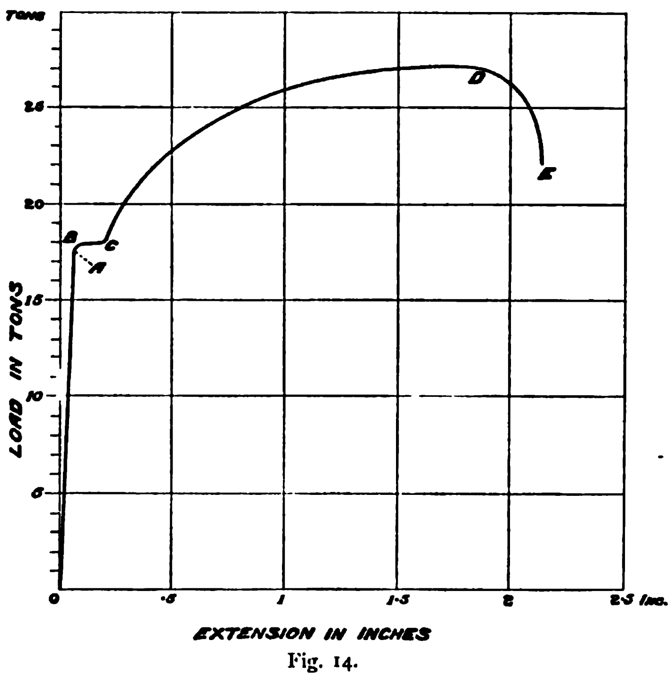


Fig. 14.

has invariably found the stress and strain diagram (drawn automatically by the bar itself) to have the characteristics illustrated by Fig. 14. From O to A the extensions of the bar are very nearly proportionate to the stress applied; in other words, they follow Hook's law *ut tensio sic vis.* A is the true elastic limit. From A to B is a transition stage;

the extension is no longer proportionate, but increases more and more rapidly. The extension between O and A is a very minute portion of the length of the bar, and is exaggerated in the diagram so as to make it capable of representation. When the stress reaches the amount indicated by the point B, the bar extends without increase of load a distance of $\frac{1}{8}$ inch or more in a specimen 10 inches long—it, so to speak, "breaks down." Hence B has been termed the "breaking-down point" of the bar. The "elastic limit," as ordinarily found by the aid of a pair of dividers, may be anywhere between A and B, or even below A.

The specimen now extends from B to C without increase of load. In diagrams taken with apparatus of too sensitive a nature in the writer's opinion to be reliable, and also in diagrams taken by apparatus in which the load on the specimen is measured by the water pressure in the hydraulic cylinder of the testing machine, the line B C appears as a jagged line.

There can be little doubt, however, that these apparent fluctuations in the load in the specimen are due mainly to imperfections in the recording apparatus, owing to the rapid stretch of the specimen from B to C. With well designed apparatus in which the actual load, as measured by the dead-weight lever, is recorded, the line between B and C is found to be almost, if not quite, straight and horizontal. C is usually a very well marked point, from which the extension of the bar increases very rapidly with increasing load. At D the maximum load which the bar can sustain without immediate fracture is reached. From D to E the load on the bar materially diminishes, until the bar, having stretched to E, suddenly breaks.

The whole subject is a very interesting one, but since we are concerned not with the behaviour of metals under test, but with their use in hydraulic machines simply, we must be brief. Our present object is to point out that the so-called "elastic limit" is not in itself a quantity of much importance,

since it can be raised at pleasure. For instance, if the bar, the behaviour of which under test is illustrated by Fig. 14, had been subjected to a preliminary load of 22½ tons, we know by the results of many experiments that, on being subsequently tested, its "elastic limit," instead of being about ·18 tons per square inch, would have been found to be more than 22½ tons to the square inch, and no such stage as that between B and C would be observed. Hence a steel or iron master, who has to do with an engineer who has great faith in a high "elastic limit" as a measure of the strength of a bar—and there are such engineers—has merely to watch his opportunity and apply a stress equal to the prescribed "elastic limit" before the inspector commences his test, and he will be sure of the bar passing the test as far as regards the "elastic limit."

Not only can the "elastic limit" be raised; it can also be lowered by manipulation. By compressing a bar of wrought iron endways, powerfully, its "elastic limit" may be reduced to as little as 5 tons per square inch without affecting sensibly its ultimate breaking weight.

Hence we must discard the "elastic limit," at any rate taken by itself, as in any way measuring the value of the bar for constructive purposes. In comparing the quality of two bars, it is necessary that the specimens should be of equal length and equal diameter. The important points, then, to be observed, as determined by tests, are the ultimate breaking weight and the ultimate extension.

Having thus disposed of the claims of the "elastic limit" to be considered as a basis from which to determine the relation between test straps and working stress, we have next to consider from what sound basis their relations may be determined in the special case which we have to consider, viz., that of hydraulic power machinery.

CHAPTER V.

TEST LOAD.

HAVING disposed of the pretension of the so-called "elastic limit" to be considered an indication of the safe working load of a bar of wrought-iron or steel, we have now to point out another fallacy, which has a deep root in the minds of many. It is a common belief that if a piece of metal or a machine pass its "test" without giving signs of undue strain by taking permanent set—for instance, in the case of a bar stressed in tension, or, as in the case of a hook or a punching machine, by a permanent springing open of the jaw—that it is quite safe for any number of repetitions of the test load. Some early experiments of Sir Wm. Fairbairn went to show the fallacy of this error in the case of riveted girders, but were too crudely conducted to be conclusive. More recently, however, the researches of Wöhler and Spangenberg have thrown a flood of light on the subject.

It appears from their experiments that the breaking weight of a piece of metal depends not merely on the absolute magnitude of the stress per square inch, but also on the frequency of repetition and the range of variation of the stress. The experiments, though very extensive and amply conclusive as to the general results, were not conducted with sufficient care to suggest an exact formula; but the general nature of the results will be readily understood by considering the breaking weights, as determined from them, of a bar of wrought iron loaded either by (1) a steady load applied constantly; (2) a steady load applied and removed alternately an indefinite number of times; (3) a steady load applied

alternately in opposite directions—that is, alternately compressing and extending the fibres.

The breaking weight in the first case is 20 tons per square inch, in the second $13\frac{1}{3}$ tons, and in the third $6\frac{2}{3}$ tons per square inch. Thus the breaking weight in the three cases have the proportions $3 : 2 : 1$, or $1 : \frac{2}{3} : \frac{1}{3}$.

As an example of the first case, we may instance the links which connect the balance-weight chain of a slow-moving hoist to the cage or to the balance weight. As an example of the second, the columns, head, cylinder, etc., of a hydraulic press; and as an example of the third, the piston rod of a steam engine, or the spindle of an overhead pulley of a hoist.

What is known as the Dynamic Theory of Loads is now largely accepted by leading engineers, more especially in connection with bridge design. The theory states that if a load be applied quite suddenly the strain produced is double of that which would result from the application of the same load very gradually; also if a load be suddenly removed, and applied in the opposite sense, the resulting strain is three times that which would result from the removal of the load and application of the reverse load very gradually.

In treating of the safe working loads, as determined from the test stress, we shall in all that follows suppose that the metal is stressed in one direction only, but that the stress is applied and removed continually in the ordinary working of the machine. If the stress be alternately applied in opposite directions, one-half the working load, as determined by the following considerations, must be taken as the safe working load.

We may divide working loads roughly into four classes—(1) Perfectly steady loads; (2) ordinary loads, not perfectly steady, but nearly so, and perfectly steady loads applied to machines in which failure would involve considerable loss or annoyance; (3) loads applied with more or less but not excessive shock; (4) loads in which failure must result in danger to life or limb.

As types of the first class of loads may be taken hand-worked hydraulic presses operating on yielding materials. Here we have the class of stress most favourable to the life of the machine, and the working stress may be four-fifths the test stress. Hydraulic punching bears and hydraulic jacks, and similar small tools will also fall under this head, and may be worked up to four-fifths their test stress if otherwise properly proportioned. Indeed, machines of this class are often worked up to their full test load. As types of the second class may be taken large hydraulic baling presses worked rapidly and frequently, high-pressure hydraulic accumulators, fitted with safety valves, and high-pressure work in general; for this class the working load may be two-thirds the test load. Medium-pressure hydraulic work, in which the load is very steady, may also be included in this class. As types of the third class may be taken medium-pressure hydraulic hoists, accumulators, etc., chain hooks and similar parts, and medium-pressure work in general, for which the working load should not exceed one-third to one-half the test load, according to the degree of shock incidental to the working of the machine. For the fourth class, which is intended to cover such work as hotel lifts, etc., the working load should not exceed from one-fourth to one-fifth the test load—abundant strength being specially provided in all parts liable to deterioration or wear. If frequent skilled supervision cannot be guaranteed, a still larger margin should be allowed.

Gun-metal high-pressure hand pumps may be worked up to two-thirds the test pressure. Gun-metal high-pressure pumps driven by steam cylinders direct, or by belt, may be worked up to half the test pressure, or, if of cast iron, up to one-third the test pressure.

Table I. gives the test stresses and working stresses suitable for the materials most frequently used in hydraulic machinery, and the proper proportion of working load to test load.

TABLE I.—STRESSES IN HYDRAULIC MACHINERY FOR LOADS APPLIED IN ONE DIRECTION ONLY.

[*With best material and workmanship the higher values may be used.*]	Cast Iron.		Wrought Iron, Forging.			Cast Malleable not more than 1 inch thick.		Cast Steel.		Forged Steel.		Gun Metal.		Phosphor Bronze.	
	Cylinder.	Ordinary.	Cylinder.	Ordinary.	Rolled Bars.	Cylinder.	Ordinary.	Cylinder.	Ordinary.	Cylinder.	Ordinary.	Cylinder.	Ordinary.	Cylinder.	Ordinary.
Ultimate Tensile Stress - Tons	9	7.5	20	22	25	18	20	28 24	..	30	34	12	..	19	..
Test Stress - - - - Tons	4 3.5	3	10 8	9	10	8	..	11 8	..	12 10	..	4	..	7.5 7	..
Ratio with Ultimate Stress -	.44 .4	.4	.5 .4	.4	.4	.44	..	.4 .33	..	.4 .33	..	.33	..	.4 .35	..
CLASSIFICATION.	WORKING STRESSES IN TONS PER SQUARE INCH.														
No Shock.—Hydraulic Machinery worked by Hand Pump	4 3.5	3	9.5 8	7	8	8	..	10 8	..	11 10	..	4	..	7	..
Ratio with Ultimate Stress	.44 .4	.4	.47 .4	.32	.32	.44	..	.36 .33	..	.36 .33	..	.33	..	.35	..
Small Shock.—Large rapidly worked Baling Presses, Accumulators with Safety Valves, and Hydraulic Machinery working at a pressure from 1 to 4 tons per square inch	2.3	2	5.3	6	6.6	5.3	..	5.3	..	6.6	..	2.6	..	4.6	..
Ratio with Ultimate Stress	.26	.26	.26	.26	.26	.26	..	.26	..	.26	..	.26	..	.26	..
Medium Shock.—Hydraulic Goods Hoist, Cranes, Jiggers, and Accumulators for Pressures not exceeding 1,500 lbs. per square inch	1.1 to 1.7 mean 1.4	1 to 1.5 1.25	2.6 to 4 3.3	3 to 4.5 3.7	3.3 to 5 4.1	2.6 to 4 3.3	.. ..	2.6 to 4 3.3	.. ..	3.3 to 5 4.1	.. ..	1.3 to 2 1.6	.. ..	2.3 to 3.5 2.9	.. ..
Ratio with Ultimate Stress	.16	.16	.16	.16	.16	.17	..	.13	..	.13	..	.13	..	.14	..
Hotel Lifts - - - - - - -	.9	.75	2	2.25	2.5	2	..	2	..	2.25	..	1	..	1.75	..
Ratio with Ultimate Stress	.1	.1	.1	.1	.1	.11	..	.08	..	.08	..	.08	..	.08	..
Pumps worked by Power - - - -	1.2	..	..	..	..	..	..	..	..	..	..	2	..	3.5	..
Ratio with Ultimate Stress	1.13	..	..	..	..	..	..	..	..	..	..	.16	..	.17	..
Pumps worked by Hand - - - -	2.3	..	..	..	..	..	..	..	..	..	..	2.66	..	4.66	..
Ratio with Ultimate Stress	.26	..	..	..	..	..	..	..	..	..	..	.26	..	.25	..

The next point to be examined is a peculiar description of stress which is only found in the thick cylinders of high-pressure hydraulic work. Fig. 15 represents a cross section through a thick cylinder. Internal radius $= r$, external radius $= r + t$ when unstressed. These radii become, when stressed by internal fluid pressure, say r_1 and $r_1 + t_1$ respectively. If the stretch of the material follow the elastic or Hooke's laws, the circumferential tension of any ring of fibres will be proportional to the whole extension of the ring

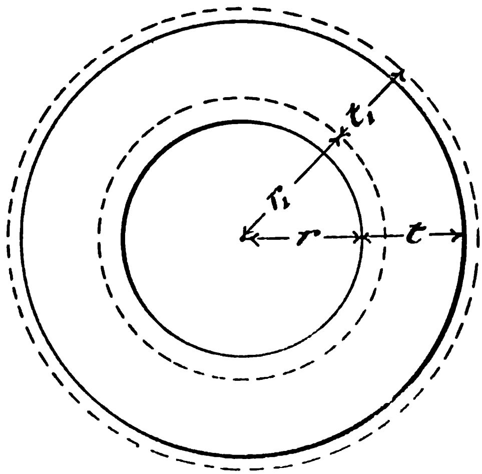


Fig. 15.

divided by the whole circumference of the ring. In other words, the tension will vary as the extension per unit of length. Hence the stress at the internal circumference of the cylinder will be to the stress at the external circumference as—

$$\frac{r_1 - r}{r} : \frac{(r_1 + t_1) - (r + t)}{r + t}$$

and since the internal pressure tends to compress the material radially, and thus cause a reduction in the thick-

ness, and as the circumferential tension also tends to reduce the thickness, t_1 is necessarily less than t, and the fraction $\frac{(r_1 - r) - (t - t_1)}{r + t}$ less than the fraction $\frac{r_1 - r}{r}$. Hence the tension on the fibres of the external circumference is less than that on those of the internal circumference, and the former do not take their fair proportion of the work of resisting the disruptive effect of the internal pressure.* Lamé was the first writer to accurately determine the effect of this inequality of stress throughout the thickness of the cylinder on the supposition of extension being directly proportional to stress. He obtained the formula $f = \frac{P(R^2 + r^2)}{R^2 - r^2}$, where f is the tension at the internal circumference, P the internal pressure, R the external radius, and r the internal radius. We have omitted from the formula the term involving the external pressure, since, in such cases as we are concerned with, the external pressure will, in general, be comparatively very small. The steps by which this result is arrived at may be consulted in Lamé's "Traité de l'Elasticité," or Ibbetson's "Theory of Elasticity," or Rankine's "Applied Mechanics," the result obtained being the same in each.

The formula may also be put in this form : $f = \frac{P}{T} \cdot \frac{R^2 + r^2}{R + r}$ where T is the thickness of the cylinder, P and f may be taken in tons or pounds per square inch, and R, r and T in inches, or any other units of length or weight at pleasure, provided the same units be used for P as for f, and the same unit for T as for R and r.

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* The above must not be taken as an exact statement of the true conditions of stress and strain throughout the metal of the cylinder, as we have not taken account of the effect of the radial compression on the relations of stress and strain, but simply as an approximate illustration of the necessary variation of the strain throughout this thickness.

If R be nearly equal to r, we obtain the usual formula for the tension on the metal of a thin cylinder, viz.,

$$f = \frac{P.r}{T}$$

Professor Pearson (see footnotes pp. 550 and 552 of Todhunter's "History of Elasticity") considers that Lamé's formula for the strength of a thick cylinder errs on the side of assigning too high a value to the strength of the cylinder. The author does not, however, consider this conclusion to be confirmed by experience. On the contrary, we know that the actual materials in construction do not follow Hooke's law in their extension with precision, and there is, so to speak, a sort of "give-and-take" action, which tends to cause a greater equality of stress throughout the thickness of a cylinder than Lamé's formula would indicate. On the other hand, however, the internal circumference of the cylinder in the case of castings is usually the most unsound, owing to the exterior of the cylinder cooling first, and the inner rings of metal later, while at the same time it is the part most severely stressed in actual work.

The plan of circulating water through the core bar, as adopted in America in the casting of ordnance, may be employed with advantage in the case of important hydraulic cylinders, to ensure soundness in the inner layers of cast-iron cylinders.

On the whole, the author considers it better to be guided by the results of successful practice in assigning the test pressure for hydraulic cylinders, rather than by a formula based on a defective theory. Tables I. and II. exemplify his own practice, and have been used successfully in fixing the dimensions of many hundreds of hydraulic cylinders. For low-pressure work, the following dimensions may be adopted for pressure (test) not exceeding 500 lbs. per square inch :—

Inside diameter in inches	3	3½	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	20	22	24	26	28	30
Thickness in inches	⅜	$\frac{7}{16}$	½	½	½	$\frac{9}{16}$	$\frac{9}{16}$	⅝	⅝	¾	¾	¾	¾	⅞	⅞	⅞	⅞	1	1	1	1	1⅛	1⅛

TABLE II.

THICKNESS IN INCHES OF CAST-IRON CYLINDERS FOR TEST PRESSURES OF

Inside Diam.	Lbs. per Square Inch.				Tons per Square Inch.								
Ins.	800	1 000	1 200	1.500	1	1 1/4	1 1/2	1 3/4	2	2 1/4	2 1/2	2 3/4	3
3	3/8	3/8	1/2	1/2	5/8	3/4	7/8	1 1/8	1 1/4	1 1/2	1 5/8	1 7/8	2
3 1/2	7/16	1/2	1/2	1/2	3/4	7/8	1	1 1/4	1 3/8	1 5/8	1 7/8	2 1/8	2 1/4
4	1/2	1/2	1/2	9/16	7/8	1	1 1/8	1 3/8	1 1/2	1 7/8	2 1/8	2 3/8	2 5/8
5	1/2	9/16	5/8	3/4	1	1 1/4	1 3/8	1 5/8	1 7/8	2 1/4	2 1/2	2 7/8	3 1/8
6	1/2	5/8	5/8	3/4	1 1/8	1 3/8	1 5/8	2	2 1/4	2 5/8	3	3 3/8	3 3/4
7	9/16	5/8	3/4	7/8	1 1/4	1 5/8	1 7/8	2 1/4	2 1/2	3	3 1/2	3 7/8	4 1/4
8	9/16	3/4	7/8	1	1 3/8	1 3/4	2	2 1/2	2 7/8	3 3/8	3 7/8	4 3/8	4 7/8
9	5/8	3/4	7/8	1 1/8	1 1/2	1 7/8	2 1/4	2 3/4	3 1/4	3 7/8	4 3/8	4 7/8	5 3/8
10	3/4	7/8	1	1 1/8	1 5/8	2 1/8	2 1/2	3	3 1/2	4 1/4	4 3/4	5 3/8	6
11	3/4	7/8	1	1 1/4	1 3/4	2 1/4	2 3/4	3 3/8	3 7/8	4 5/8	5 1/4	6	6 5/8
12	7/8	1	1 1/8	1 3/8	2	2 1/2	3	3 5/8	4 1/4	5	5 3/4	6 1/2	7 1/8
13	7/8	1	1 1/4	1 1/2	2 1/8	2 3/4	3 1/4	3 7/8	4 1/2	5 3/8	6 1/8	7	7 3/4
14	7/8	1 1/8	1 1/4	1 1/2	2 1/4	2 7/8	3 3/8	4 1/8	4 7/8	5 3/4	6 5/8	7 1/2	8 1/4
15	1	1 1/8	1 3/8	1 5/8	2 3/8	3	3 5/8	4 1/2	5 1/4	6 1/8	7 1/8	8	8 7/8
16	1	1 1/4	1 1/2	1 3/4	2 1/2	3 1/4	3 7/8	4 3/4	5 1/2	6 1/2	7 1/2	8 1/2	9 1/2
17	1 1/8	1 1/4	1 1/2	1 7/8	2 5/8	3 3/8	4 1/8	5	5 7/8	7	8	9	10
18	1 1/8	1 3/8	1 5/8	2	2 3/4	3 5/8	4 3/8	5 3/8	6 1/4	7 3/8	8 3/8	9 1/2	10 1/2
20	1 1/4	1 1/2	1 3/4	2 1/8	3	3 7/8	4 3/4	5 7/8	6 7/8	8 1/8	9 3/8	10 5/8	11 3/4
22	1 3/8	1 3/8	1 7/8	2 3/8	3 3/8	4 3/8	5 1/4	6 3/8	7 1/2	8 7/8	10 1/4	11 5/8	12 7/8
24	1 3/8	1 5/8	2 1/8	2 1/2	3 5/8	4 3/4	5 7/8	7	8 1/4	9 3/4	11 1/8	12 5/8	14
26	1 1/2	1 7/8	2 1/4	2 3/4	3 7/8	5 1/8	6 1/4	7 5/8	8 7/8	10 1/2	12	13 5/8	15 1/8
28	1 5/8	2	2 3/8	2 7/8	4 1/4	5 1/2	6 5/8	8 1/8	9 1/2	11 1/4	13	14 5/8	16 1/4
30	1 3/4	2 1/8	2 5/8	3 1/8	4 1/2	5 7/8	7 1/8	8 3/4	10 1/4	12 1/8	13 7/8	15 3/8	17 3/8

TABLE III.

Thickness of Steel Cylinders (Unhammered Castings) for Test Pressures of

Inside Diam. Ins.	Tons per Square Inch.											
	1	1¼	1½	1¾	2	2¼	2½	2¾	3	3½	4	5
3	...	...	...	...	...	...	...	...	...	1	1⅛	1¼
3½	...	...	...	...	...	...	...	...	...	1⅛	1¼	1½
4	...	...	...	...	...	...	...	...	1	1¼	1⅜	1⅝
5	...	...	...	...	...	...	1⅛	1¼	1¼	1½	1⅝	2
6	...	...	...	...	1	1⅛	1¼	1⅜	1⅜	1⅝	1⅞	2¼
7	...	...	...	1	1⅛	1¼	1⅜	1½	1⅝	1⅞	2⅛	2⅝
8	...	...	1	1⅛	1¼	1⅜	1½	1⅝	1¾	2⅛	2⅜	3
9	...	1	1⅛	1¼	1⅜	1⅝	1¾	1⅞	2	2⅜	2⅝	3¼
10	...	1⅛	1¼	1⅜	1½	1¾	1⅞	2	2⅛	2½	2⅞	3⅝
11	1	1¼	1⅜	1½	1⅝	1⅞	2	2¼	2⅜	2¾	3⅛	4
12	1	1¼	1⅜	1⅝	1¾	2	2⅛	2⅜	2½	3	3⅜	4¼
13	1⅛	1⅜	1½	1¾	1⅞	2⅛	2⅜	2⅝	2¾	3¼	3¾	4⅝
14	1⅛	1⅜	1⅝	1⅞	2	2¼	2½	2¾	2⅞	3½	4	5
15	1¼	1½	1¾	2	2⅛	2⅜	2⅝	2⅞	3⅛	3¾	4¼	5¼
16	1¼	1½	1¾	2	2¼	2½	2¾	3	3¼	3⅞	4½	5⅝
17	1⅜	1⅝	1⅞	2⅛	2⅜	2¾	3	3¼	3½	4⅛	4¾	6
18	1⅜	1¾	2	2¼	2½	2⅞	3⅛	3⅜	3⅝	4⅜	5	6¼
20	1½	1⅞	2⅛	2½	2¾	3⅛	3⅜	3¾	4	4¾	5½	7
22	1⅝	2	2⅜	2¾	3	3⅜	3⅝	4	4⅜	5¼	6	7⅝
24	1¾	2⅛	2½	2⅞	3¼	3⅝	4	4⅜	4¾	5⅝	6½	8¼
26	1⅞	2⅜	2¾	3⅛	3½	4	4⅜	4¾	5⅛	6⅛	7⅛	9
28	2	2½	2⅞	3⅜	3¾	4¼	4⅝	5⅛	5½	6⅝	7⅝	9⅝
30	2⅛	2⅝	3⅛	3⅝	4	4½	5	5½	5⅞	7	8⅛	10¼

A few remarks may be here appropriately introduced on certain points in the design and construction of high-pressure hydraulic cylinders of these materials, non-attention to which will frequently result in failure and disappointment.

In the first place, the internal corners at the bottom should be struck to a large radius, as shown by Fig. 16; and if the cylinder be cast with a solid bottom, the interior of the bottom should be struck to a radius not exceeding the diameter of the cylinder in length. A good practical rule is to make the corners one-fourth the internal diameter of the

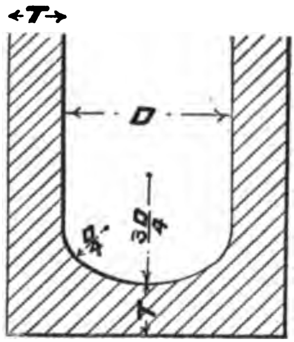


Fig. 16.

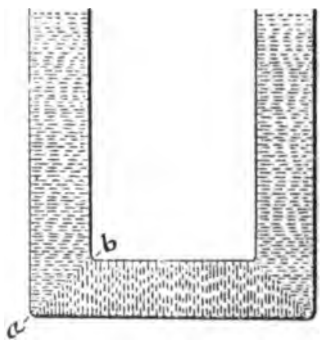


Fig. 17.

cylinder in radius, and the bottom three-fourths the internal diameter of the cylinder in radius. If these proportions be adopted, the thickness of the bottom of the cylinder will be sufficient if made equal to that of the walls, as illustrated by Fig. 16. In the case of long cylinders, in which it is necessary to carry the core bar through the bottom in order to provide a support for its end, the same proportions may be adopted, simply inserting the necessary plug for stopping the hole left by the core bar.

The necessity of a large rounding of the corners arises from the fact that if they be left nearly square (see *b*, Fig. 17),

the crystals of the casting arrange themselves during cooling in such a manner as to invite fracture along the line *a b* (Fig. 17), and unless the cylinder be constructed of a thickness unnecessarily great for the pressure to which it is subjected, deterioration gradually goes on along the line *a b*, until sooner or later failure takes place, as illustrated by Fig. 18; and a conical piece A breaks away from the end of the cylinder. Fig. 19 shows the arrangement of crystals in a cylinder with a curved bottom of equal thickness to the sides.

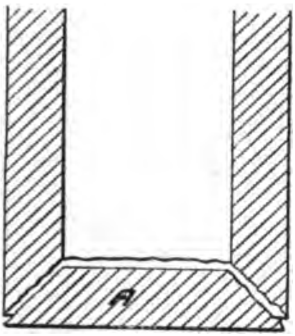


Fig. 18.

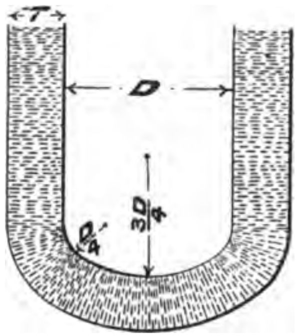


Fig. 19.

Fig. 20 illustrates a properly-designed cylinder, and similar to Fig. 16, but with a plug inserted by driving from the inside. This method is found amply sufficient for cylinders of diameters ranging to 10 inches or 12 inches inside, or even more. For larger cylinders, the method illustrated by Fig. 21 may be adopted, in which the plug is made tight by means of a U leather and back plate.

The sources of weakness to which attention was drawn in Chapter III. should also be carefully avoided, and it is also in general advisable to construct high-pressure hydraulic cylinders in the form of plain cylinders, as the castings are

depth to produce adequate fluid pressure on the casting, but also of sufficient *bulk*, in order that it may remain fluid longer than the body of the cylinder, and thus maintain a pressure on the metal during the whole period of solidification. Hence, to be effective, the head should take the form illustrated by Fig. 22 or 23, and not that illustrated by Fig. 24, which is ineffective and irrational, though not unfrequently adopted.

If due attention be paid to the points here briefly discussed, the thicknesses given in Tables II. and III. will be found amply sufficient for the test pressures there stated.

Having thus cleared the ground by defining the meaning to be assigned to tests and working pressure and stress, and their proper relative and absolute values for the various materials employed in the construction of hydraulic machinery, we are now at liberty to discuss the proper proportions and design of the details and component parts of such machinery.

PART III.—JOINTS.

CHAPTER VI.

PACKINGS FOR SLIDING SURFACES.

THE packing by means of which the rams, pistons or plungers of hydraulic machinery are enabled to slide to and fro at the same time that the passage of fluid past the sliding surfaces is prevented, may be divided into two classes, viz., firstly, that in which the packing is self-acting—that is,

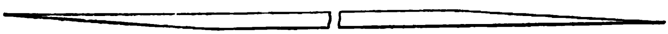


Fig. 25.

maintained in water-tight contact with the sliding surface by the simple action of the hydraulic pressure itself; and secondly, that in which the tightness of the packing is dependent on mechanical compression by means of glands or junk rings, as in the case of stuffing boxes.

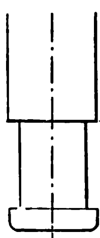


Fig. 26.

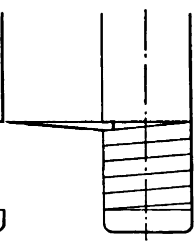


Fig. 27.

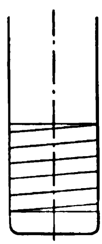
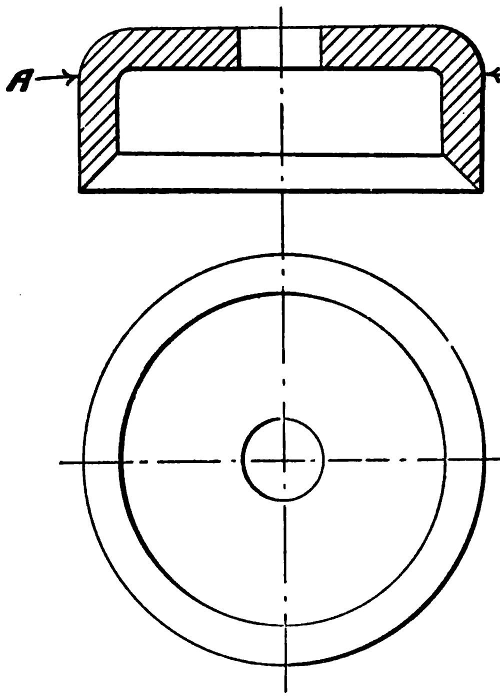


Fig. 28.

Of the first, or self-acting class of packing, the simplest is the spiral leather packing (Figs. 25, 26, 27, and 28). This is a very excellent packing for small plungers and pistons. It consists simply of a strip of supple leather $\frac{3}{16}$ inch or

¼ inch wide, and of sufficient length to wrap round the plunger three, four, or five times (Fig. 25). Fig. 26 represents the plunger without the packing, Fig. 27 the packing in course of being wound on, and Fig. 28 the plunger packed and ready for use. The operation of packing a



Figs. 29 and 30.

plunger in this manner is apparently very simple, but yet requires a certain amount of skill and practice to perform it with speed and neatness. The strip of leather must first have one end cut with a sharp knife to an acute angle. It must then be tried in the groove of the plunger, and shaved

if necessary down to the proper thickness to just fill the groove up to the required working diameter which will fit the pump barrel tightly. It is then wrapped round the plunger, and the free end chamfered off to a gradual taper and length to just fill the length of the groove. The free end is then hammered into the unfilled portion of the groove with the handle of a screwdriver or file, and the plunger is ready for use.

This description of packing is only suitable for small plungers not exceeding 1 inch or $1\frac{1}{2}$ inches diameter, but is

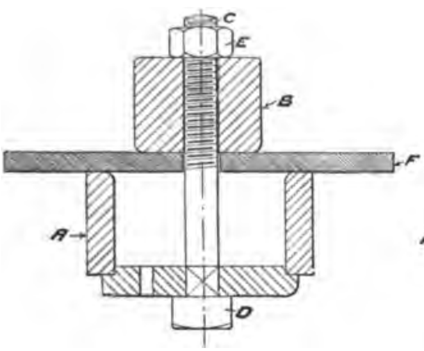


Fig. 31.

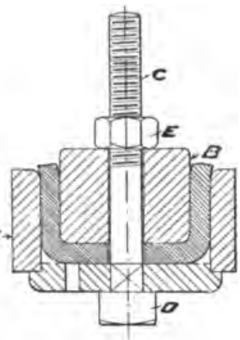


Fig. 32.

a very simple, cheap and durable packing for such small work, and is perfectly reliable and water-tight at even the highest pressures.

The most simple self-acting packing for rams, pistons and plungers, next to the spiral leather packing previously described, is the cup type of packing, which is constructed in three forms, commonly termed cup, hat and U packing respectively. The cup packing is illustrated by Figs. 29 and 30, and simple tools for and the process of manufacture by Figs. 31 and 32.

The cup packing is used as a packing for pistons, for

making water-tight joints at the ends of plugs and plungers, and similar purposes, and owes its self acting tightness to the pressure of the water on the internal surface of the cup, which expands the rim of the cup and forces it against the pump barrel or other surface with which water-tight connection is to be maintained. It might at first sight appear that the whole depth of the cup would be directly useful in forming the joint; or, in other words, that the hydraulic pressure acting on the internal surface of the rim of the cup would press the whole external surface of the rim of the cup against the pump barrel, and that hence the water-tightness of the packing would be enhanced by increasing the depth

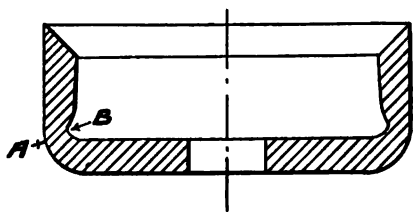


Fig. 33.

of the cup rim. This is not, however, found to be the case in practice. The effective portion of the cup is merely a narrow ring of surface near the point A, Fig. 29, where the leather touches the piston, and the remaining portion of the cup leather is in a great measure superfluous. This fact is evidenced in several ways in a very convincing manner. For instance, the wear takes place almost entirely at A.

Fig. 33 represents a section through a worn-out packing. The indentation B inside the packing is due to the external wear of the packing at A, as the leather is forced out by the internal pressure from the inside of the cup to supply the portions worn away by external friction. The localisation of the wear is so marked as to lead superficial observers to

suppose that the leather has been cut by the pressure of the edge of the piston. The effect is, however, entirely due to fair wear, and is not to be obviated by rounding the edge of the piston or other such expedients occasionally suggested.

Another proof is furnished by the fact that the friction of the cup is independent of the depth of the rim, and is the same practically for a packing 2 inches deep as for one an inch or less in depth; whereas, were the water-tightness of the cup due to the pressure on the whole internal surface of the rim, it would be reasonable to suppose that the friction would increase with the depth of the cup.

The manufacture of a cup leather is a very simple operation. A disc F of leather (see Fig. 31) of suitable diameter is soaked in warm water until quite pliable. It is then placed centrally on the hollow mould A, and the plunger B screwed down on it by means of the central screw C (the head D of which may be conveniently held in a vice) and nut E, until it is forced into the mould A. When the leather is dry the edge is trimmed off to an angle of 45°, either by means of a sharp knife, or, preferably, in a wood chuck in the lathe. If the leather is required without a central hole, external clamps may be used in place of the central screw C to force the plunger B into the mould. If a number of leathers are to be manufactured, a small hydraulic press, of about 10 tons power, will be found very convenient, as also a sheet-iron oven heated by steam for drying the packings. The latter, however, requires great care in use, as, if overheated in drying, the leathers rapidly fail in ordinary work. It is poor economy to use inferior material for hydraulic leathers. Sound oak-tanned leather should be selected, cut from the best part of the butt. If the packings are not subject to much wear, indiarubber cups may, however, be used with advantage in all cases where packings are liable to become dry through being used only occasionally.

It has been previously remarked that the depth of a cup packing has but little influence on its water-tightness. We

may further add that it is really prejudicial to the efficiency and durability of the packing to make the rim of the leather unduly deep, for the simple reason that the stress on the leather during its manufacture is greatly increased by increasing the depth of the cup. This stress is greatest also at the very part (A) of the leather which is subject to the greatest wear in actual work. If the cup be deep, and very great care be not taken in the manufacture, the leather is liable to tear at this point, or, if not actually torn, to suffer great deterioration, which, although it may be disguised and concealed by subsequent dexterous manipulation, never fails to show itself afterwards in an abnormally short life of the leather. There is no advantage whatever in making the cup more than 1 inch deep, and any greater depth than this is not merely useless, but, for the reason here pointed out, really undesirable as leading to injury to the packing at the very part at which the greatest soundness is required.

The barrel in which the cup leather works should, if possible, be lined with gun metal or brass. For medium and high pressures it should invariably be so lined. The attempt to use leather packings under high pressures for pistons working in cast-iron barrels, unlined, always results in great annoyance and frequent delays from the rapid deterioration of the bore of the cylinder, and consequent constant failure of the packings, which are only durable when they have an absolutely smooth surface unaffected by corrosion to work against. In the case of thick cast-iron cylinders working at high pressures, owing, apparently, to the comparative porosity or looseness of texture of the interior surface of the casting forming the bore, which has already been commented on, the friction of the leathers appears at times to tear away considerable portions of the internal surface, leaving rough places, which destroy the packings after a few passages over them. Steel castings are not free from this defect, and suffer occasionally even more than cast iron.

These remarks do not, however, apply so strongly to cast-

iron rams, the external surface of which is generally very close in texture and capable of receiving a high polish, and can also be readily kept in good condition as regards polish and lubrication. Even in the case of rams, however, it has been found highly conducive to the durability of the leathers to case the lower part of the rams of hydraulic presses, for instance, with gun metal. The rams of hydraulic presses for baling Manchester goods, and for cotton pressing, are invariably so cased by first-class makers.

The laws governing the friction of cup and similar leathers were investigated carefully by Mr Hick, of Bolton, and found to be in the main very simple. The author's own experience fully endorses Mr Hick's results, which may be stated in the following form :—

Let P be the total load on a ram or piston, and D its diameter in inches. The whole friction of the packing of the ram or piston is—

$$f = \frac{4P}{100D} = .04\,\frac{P}{D},$$

the leather packing being in the condition as regards lubrication usually met with in practice, and the ram and cylinder in first-class condition as regards polish and soundness of surface. For instance, let the ram of a press be 10 inches diameter and the load be 100 tons, corresponding to a hydraulic pressure of 1.27 tons per square inch, then the friction of the packing will be—

$$f = \frac{4 \times 100}{100 \times 10} = .4 \text{ tons} = 8 \text{ cwt.};$$

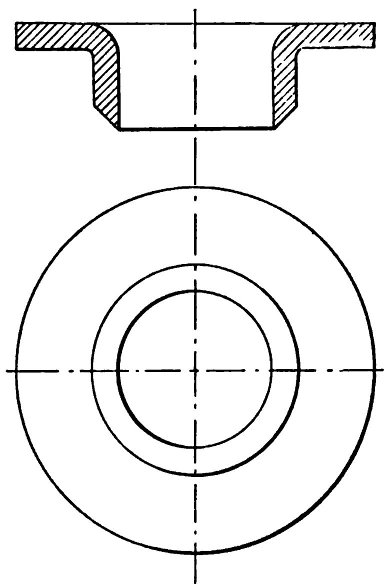
or $\frac{4}{10}$ per cent. of the whole load. The friction in this case is a very inconsiderable amount compared with the total load, but if the packing be small in diameter the percentage of the whole pressure absorbed by friction becomes very appreciable, and must be taken carefully into account when

designing apparatus involving the use of pistons or plungers packed with leather for determining the intensity of hydraulic pressures.

For instance, if the packing be $\frac{1}{4}$ inch in diameter, the percentage of the whole load absorbed by the friction of the packing will be—

$$4 \div \tfrac{1}{4} = 16 \text{ per cent.},$$

which is a very notable amount.

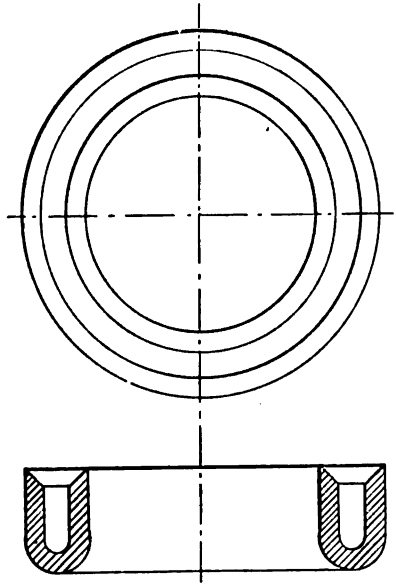


Figs. 34 and 35.

It will be observed that the above remarks as to the friction and wear and tear of leather packings apply equally to all leather packings of the cup type, and not merely to cups, but also to hat and U packings.

The action of the hat packing (Figs. 34 and 35) and U

packing (Figs. 36 and 37) is, indeed, identical with that of the cup packing proper. The point of greatest wear and the method of calculating the friction are the same for all three kinds of packing. The tools used in and mode of manufacture are, however, different, for neither the hat packing nor the U can be made in so simple a manner. Figs. 38



Figs. 36 and 37.

and 39 illustrate the formation of the hat packing from a circular disc of leather. The packing is finished by cutting out the central disc and chamfering the edge to an angle of 45°.

The pressure employed in forcing the leather into the die may be supplied by means of a central screw and nut, as previously described for the ordinary cup packing (p. 69).

In this case, of course, a small hole must be first cut in the disc of leather for the central screw to pass through. This hole must in any case be small, otherwise it will be found impossible to make a satisfactory packing on account of the tearing and distorting of the leather. If screw clamps, or a small screw, or hydraulic press be employed, however, the central hole may be dispensed with. These remarks apply equally to the manufacture of U leathers, which indeed are frequently made by means of the press in which they are subsequently to be used.

The dies used in the production of U leathers are illustrated by Figs. 40 and 41.

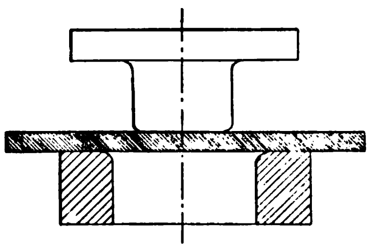


Fig. 38.

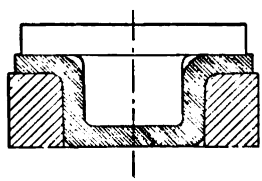


Fig. 39.

The pressing is effected in two stages; first the leather is pressed into a cup shape (see Fig. 40); and at a second operation (Fig. 41) the cup is pressed into a hat shape, with a U-shaped rim, part of the rim of the original cup going to form the internal rim of the U, as will be readily understood from the figures. The central disc is then cut out and the edges chamfered to an angle of 45°, as in the case of the hat packing.

The discs of leather used in the manufacture of leather packings are very readily and rapidly cut out of the hide by means of a knife-cutter fitted to the end of an ordinary hand-drill, and adjustable to any radius by a set screw, the discs

cut out of the centre of large packings being, of course, used for smaller packings.

The formula which we have already given for the friction of cup, U and hat packings, viz.,

$$f = .04 \frac{P}{D}$$

where f is the friction of the leather packing, P the whole load on the ram or piston, and D its diameter in inches, may be conveniently thrown into a form in which the friction is given as a function of the hydraulic pressure per square inch and diameter of the packing. For if p be the pressure per square inch—

$$P = pD^2 \times .7854 \text{; and hence}$$
$$f = .04 \times .7854 \times pD = .0314 \times Dp.$$

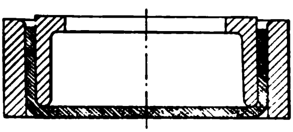


Fig. 40.

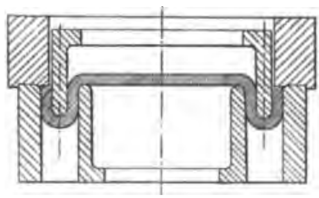


Fig. 41.

In this form the formula is applicable to packings used for other purposes than maintaining rams or pistons tight, the pressure per square inch and diameter of the packing alone being required to be known.

From the foregoing brief description of the method of working leather hydraulic packings, the truth of our remarks as to the inadvisability of employing an unnecessarily deep packing will be sufficiently apparent, especially as regards U packings. Fig. 42 illustrates the proportions to be recommended for ordinary U packings, which will indeed be found ample for all purposes. The internal diameter of a U packing should be about $\frac{1}{16}$ inch less than that of the ram which

passes through it, and the external diameter about $\frac{1}{16}$ inch greater than the recess or cylinder in which it fits, the diameter being measured at A and B. This will ensure the tightness of the packing when first inserted. For large

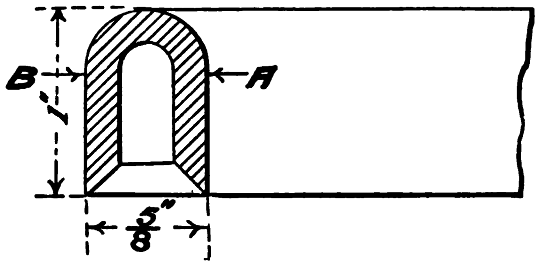


Fig. 42.

packings a somewhat greater margin may be allowed. It is always best to fit the mouths of cylinders in which U leathers are used with glands (Fig. 43), the mouth of the ram being well rounded, so that the leather can be put in place without any injury to its shape or edges. The ends of rams should

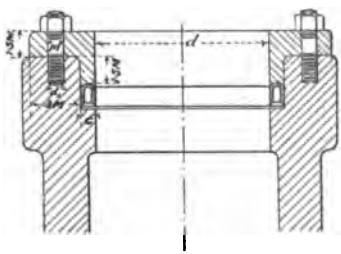


Fig. 43.

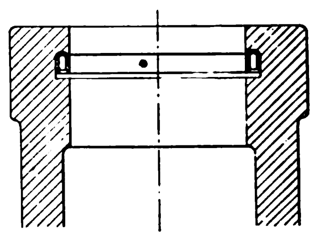


Fig. 44.

similarly be well rounded or tapered for a distance of say half an inch, with the same object.

For many purposes it is, however, sufficient to simply turn a groove in the mouth of the cylinder to receive the packing,

as in Fig. 44. The leather, if of large diameter, is easily inserted in the groove by first doubling it into the shape illustrated by Fig. 45, but, if small, practice and care are necessary to avoid injury to the leather. A small leather is usually inserted by first suppling it by letting oil stand in the rim a short time, if the leather be at all harsh; it is then pushed into the groove as far as it can be got to go, leaving as little remaining out of the groove as possible, and a blow or two from a piece of wood struck by a hammer will then usually suffice to put it in the shape illustrated by Fig. 46, and another blow at H will drive it neatly into the groove.

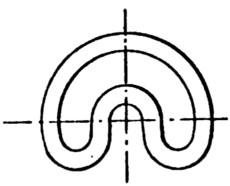


Fig. 45.

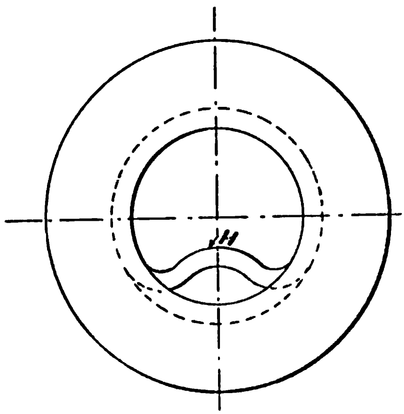


Fig. 46.

It is, however, better practice to fit the mouth of the cylinder with a gland. The studs securing the gland should not be subjected to a test stress exceeding 5 tons per square inch, if of wrought iron, and if this maximum be not exceeded, a sufficient margin of strength will be provided to compensate for extra stresses due to unequal tightening of the nuts. The thickness of the flange of the gland, if of cast iron, may be $1\frac{1}{2}$ times the diameter of the studs, and

the width of the flange three times the diameter of a stud. The projecting portions of the gland should be 1½ times the stud in length.

If p be the hydraulic test pressure per square inch, d the diameter of the ram, and c the width of the packing, the whole stress on the studs due to the hydraulic pressure is—

$$(d+c)\,c\,\pi\,p = 3.1416\,(d+c)\,pc.$$

Hence if n be the number of studs, and d_1 the diameter of a stud at the bottom of a thread, the stress on the studs per square inch is—

$$(d+c)\,c\,\pi\,p \div n\,d_1^2\frac{\pi}{4} = \frac{4pc(d+c)}{nd_1^2}$$

which, as before stated, should not exceed 5 tons, or about 11,200 lbs.

We have recommended $\frac{5}{8}$ inch as the most suitable dimensions for c, but if circumstances render it advisable to reduce the space occupied by the packing to minimum limits, c may be diminished to $\frac{7}{16}$ inch without very greatly subtracting from the efficiency of the packing.

There is a difference of practice among manufacturers of hydraulic packing leathers, some preferring to use the grain and some the skin side of the hide for the wearing surface. The latter plan makes the neatest leather in appearance, and is generally to be recommended.

Hemp Packing.—The first cost of leather hydraulic packings is comparatively high, and if the surfaces against which they work are not carefully looked after, and maintained in a state of perfect polish and well lubricated, the packings will deteriorate rapidly and become no inconsiderable portion of the expense of maintenance of a hydraulic plant. For these reasons hemp packings, which are water-tightened by strong mechanical compression by means of a stuffing box and gland, are used by many engineers wherever possible; since

the first cost of the hemp packing is comparatively inconsiderable, while at the same time the packing can be renewed more rapidly and with less loss of time. If the rod or plunger which is to be packed is heated, as is necessarily the case with some types of steam pumps, leather packings are altogether inadmissible, and hemp, asbestos, or some similar packing must be used.

On the other hand, the friction of the mechanically compressed hemp packing is far greater than that of the self-acting leather packing; also, if the hydraulic pressure for which the packing is used be high (and hemp packing, contrary to the opinion of many, may be employed successfully for very high pressures, such as 3 tons or more per square inch), there is considerable risk of scoring the surfaces of the ram and plungers in actual work, owing to the necessarily intense pressure with which the packing must be forced against the sliding surface in order to secure water-tightness. A further objection to hemp packing is that the packing must be compressed with sufficient force to ensure its being tight under the highest pressure at which the machine in which it is used is intended to work; hence, although the machine may be frequently working under a comparatively low pressure, the friction of the packing is always that due to the higher pressure, and may amount to a very large percentage of the whole work done by the machine, whereas, if leather packings be used, since the pressure on the packing varies directly with the work which the machine is performing, the *percentage* of power absorbed by the friction of the packings is, within certain limits, practically constant.

It must be left, then, to the judgment of the engineer to decide which description of packing shall be employed in any given case, each type having its own special advantages and defects, which must be duly weighed and taken into consideration before arriving at a decision. The friction of hemp packings cannot be so definitely determined by ex-

periment for any given conditions of use as that of leather hydraulic packings. We have not merely to consider the intensity of the hydraulic pressure employed, as in the case of leathers, but the depth of the stuffing boxes and diameter of the packing surface, as also the degree of pressure applied by means of the stuffing box gland. Under the same degree of compression there is no doubt that a deep stuffing box will produce more frictional resistance than a short one; but, on the other hand, the deep stuffing box will not require so intense a compression as the short one, and hence in actual practice the friction of the short stuffing box may exceed that of the long one, if the packing is to be water-tight under a given maximum pressure. It is, however, very desirable in practice to have a simple formula by which to determine the probable maximum friction of a hemp packing under given conditions. If the packing be screwed up judiciously, and the stuffing box of fair proportions, the formula may take the form of $cpd=f$, where c is a constant, to be determined by experiment within assigned limits as to pressure and diameter, p the hydraulic pressure (maximum) per square inch, d the diameter of the ram or rod in inches, and f the total amount of the friction. For many purposes it is sufficient to take f as equal to one-tenth the pressure per square inch, multiplied by the diameter of the ram, or $f=\frac{pd}{10}$, and the friction of a hemp packing judiciously used will rarely exceed this amount within very wide limits of pressure and diameter.

A very simple method of ascertaining the approximate friction of a ram packing is available when the ram can be loaded and fixed so as to rise and fall vertically. Let the ram be loaded, perfectly centrally, with any weight, the amount of which need not be exactly ascertained, and let the pressure per square inch required to raise the ram at the lowest speed be ascertained by means of an accurate pressure gauge communicating directly with the cylinder,

and let the pressure be P_1. Next let the pressure in the cylinder be similarly ascertained when the ram is descending as slowly as possible, and let the pressure be P_2. It is very important that the motion of the ram should be exceedingly slow during the experiment. Then the friction of the packing will be approximately $\frac{P_1 - P_2}{2} \times$ area of ram in square inches.

It is most necessary in carrying out such an experiment as this, however, to test the accuracy of the pressure gauge employed, since the ordinary commercial pressure gauge is frequently grossly inaccurate, and in the case of high hydraulic pressures as a general rule absolutely unreliable.

The following table gives suitable dimensions of the packing space for stuffing boxes of various diameters:—

Diameter of Ram.	Diameter of Stuffing Box Inside.	Depth of Stuffing Box.	Diameter of Ram.	Diameter of Stuffing Box.	Depth of Stuffing Box.
Inches.	Inches.	Inches.	Inches.	Inches.	Inches.
1	$1\frac{5}{8}$	2	12	$14\frac{1}{4}$	$6\frac{1}{2}$
2	$2\frac{7}{8}$	3	14	$16\frac{3}{8}$	7
3	$4\frac{1}{8}$	$3\frac{1}{2}$	16	$18\frac{1}{2}$	$7\frac{1}{2}$
4	$5\frac{1}{4}$	4	18	$20\frac{3}{4}$	$7\frac{3}{4}$
5	$6\frac{3}{8}$	$4\frac{1}{2}$	20	$22\frac{7}{8}$	$8\frac{1}{4}$
6	$7\frac{5}{8}$	$4\frac{3}{4}$	22	25	$8\frac{1}{2}$
7	$8\frac{3}{4}$	$5\frac{1}{4}$	24	$27\frac{1}{4}$	9
8	$9\frac{7}{8}$	$5\frac{1}{2}$	26	$29\frac{1}{4}$	$9\frac{1}{4}$
9	11	$5\frac{3}{4}$	28	$31\frac{1}{2}$	$9\frac{1}{2}$
10	$12\frac{1}{8}$	6	30	$33\frac{1}{2}$	10

The dimensions of the gland studs for stuffing boxes should be proportioned in a similar manner to those for the glands for U leathers, but with a larger margin of strength.

Let, as before, n be the number of studs or bolts—

In inches:
- d_1 the diameter of a stud at the bottom of the thread.
- D the diameter of the ram or rod.
- D_1 the internal diameter of the stuffing box.
- P the maximum pressure in pounds per square inch.

Then d_1^2 should not be less than

$$\frac{(D_1 - D)(D_1 + D) P}{5000 \times n}$$

The thickness of the flange of the gland should not be less than $1\frac{3}{4}$ times the external diameter of the stud, and its width may be three times the diameter of a stud for cast iron.

P in the above formula is to be taken as the maximum working pressure, or one-half the test pressure, the larger of the two values being selected; that is, if the maximum working pressure be greater than half the test pressure, P must be taken equal to the working pressure; but if half the test pressure be greater than the maximum working pressure, then P should be taken equal to half the test pressure.

Table IV. gives the efficiencies of rams or rods, working with leather or hemp packing. It has been calculated from the preceding rules, and will be found to agree with practice, providing the stuffing box is of fair proportions, and the ram or rod polished and lubricated.

Let P = gross power of ram = area of ram multiplied by pressure per square inch.

,, P_1 = nett power of ram.

,, c = coefficient, taken from table.

Then $P_1 = cP$.

TABLE IV.

COEFFICIENTS OF RAM EFFICIENCIES FOR HEMP OR LEATHER PACKING.

Diameter of Ram.	Stuffing Box.	Leather Packing.	Diameter of Ram.	Stuffing Box.	Leather Packing.
Inches.			Inches.		
$\frac{1}{16}$	...	.36	$3\frac{1}{2}$	.96	.98
$\frac{3}{32}$	...	.57	$3\frac{3}{4}$	.96	.98
$\frac{1}{8}$	...	.68	4	.96	.99
$\frac{3}{16}$	...	.78	$4\frac{1}{2}$	.97	.99
$\frac{1}{4}$	.49	.84	5	.97	.99
$\frac{5}{16}$	.59	.87	$5\frac{1}{2}$	.97	.99
$\frac{3}{8}$	.66	.89	6	.97	.99
$\frac{7}{16}$	.70	.90	$6\frac{1}{2}$	.98	.99
$\frac{1}{2}$	.74	.92	7	.98	.99
$\frac{9}{16}$	.77	.92	$7\frac{1}{2}$	.98	.99
$\frac{5}{8}$	.79	.93	8	.98	.99
$\frac{11}{16}$	.81	.94	$8\frac{1}{2}$	.98	.99
$\frac{3}{4}$	.83	.94	9	.98	.99
$\frac{7}{8}$	.85	.95	$9\frac{1}{2}$	.98	.99
1	.87	.96	10	.98	.99
$1\frac{1}{8}$	.88	.96	11	.98	.99
$1\frac{1}{4}$	.89	.96	12	.98	.99
$1\frac{3}{8}$	.90	.97	13	.99	.99
$1\frac{1}{2}$	.91	.97	14	.99	.99
$1\frac{5}{8}$	.92	.97	15	.99	.99
$1\frac{3}{4}$	.92	.97	16	.99	.99
$1\frac{7}{8}$	.93	.97	18	.99	.99
2	.93	.98	20	.99	.99
$2\frac{1}{4}$	.94	.98	22	.99	.99
$2\frac{1}{2}$	.94	.98	24	.99	.99
$2\frac{3}{4}$	.95	.98	26	.99	.99
3	.95	.98	28	.99	.99
$3\frac{1}{4}$	.96	.98	30	.99	.99

CHAPTER VII.

PIPE JOINTS.

IN our last chapter we described the usual methods of making the joints between sliding surfaces water-tight by means of animal and vegetable packings, in a self-acting manner or by forcible mechanical compression of the packing material by means of glands or bolts, or their equivalents. In the present article we propose to treat similarly of the various methods of making the joints between surfaces, fixed with reference to each other, water-tight. The joints between such surfaces are made either by placing between them suitable sheets or rings of canvas, lead, copper, leather, indiarubber, guttapercha, paper, and various other material, and forcing them tightly together by means of bolts and nuts, or their mechanical equivalents; or by using U or similar self-acting packings. In designing such a joint we have principally to consider the stress which must be brought upon the metal of the bolts and nuts in order to ensure water-tightness under a given pressure, and the dimensions which it is advisable to give the flanges, in practice, in order that they may be of adequate strength to resist the stress thus brought upon them. The stress upon the bolts, considered as a simple tensile stress, consists of two parts in general—one due solely to the hydraulic pressure on the surface exposed to it, which may be exactly calculated when the extent of that surface is known, and the pressure per unit of area to which it is subject; and another part due to the elastic reaction of the surfaces themselves and that of the joint material between them.

To make this clear, we will consider a joint such as that

illustrated by Fig. 47, in which B may be a valve chest, for instance, and A its cover; the joint being made by truly facing the surfaces, painting them, inserting a sheet of brown paper say between them, and then drawing them forcibly together by screwing up the nuts and bolts which pass through the flanges. If the nuts be screwed up when pressure is not admitted to the valve chest B, a complicated stress is brought upon the metal of the bolts— mainly a longitudinal tension, but complicated by torsional stress due to the inclination of the helix of the screw-thread and the friction between the thread and nut brought into play by the twisting action of the spanner, and complicated in addition

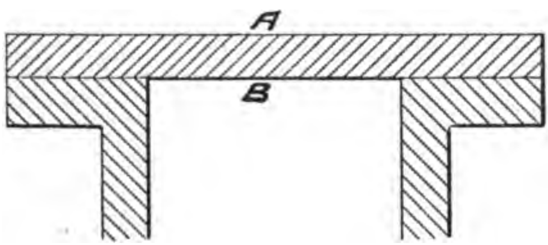


Fig. 47.

by possible bending stresses due to inequality or unequal yielding of the joint surfaces and flanges. For true surfaces and faced nuts, we may, however, treat the stress in practice as a simple tension. Let L be the length of the spanner used in inches, and F the force in pounds applied at its end by the workman in screwing up; then for ordinary bolts, having Whitworth threads, the total stress in tension on the metal of the bolt may be fairly taken at an average value of

$$T = \frac{6FL}{d} \text{ in pounds,}$$

where T is the whole stress on the bolt in pounds, considered as tensile, and d is the diameter of the bolt over the

thread in inches, the stress on the bolt per square inch at the bottom of the thread may, of course, be found by dividing T by the area of the section at the bottom of the thread.

If now water be admitted to the valve box B, at a pressure of p pounds per square inch, and S be the surface of the cover A exposed to the pressure in square inches, the whole upward pressure on the cover A will be pS in pounds, and this pressure may be transmitted to the bolts practically undiminished or increased, in addition to the stress T due to the screwing up, making the whole load on the bolts

$$= nT + pS = \frac{6FLn}{d} + pS,$$

where n is the number of bolts.

We say *may* be so transmitted advisedly, as the determination of the exact amount which will be added to the initial stress on the bolts in every particular case is highly complex, and indeed hopeless from an engineer's point of view, in very many cases depending, as it does, on the extensibility or compressibility of the various parts forming the joint. In practice we need not, however, enter into such an investigation; it is sufficient for our purpose to know that the whole load on the bolts of the joint is not likely to exceed the amount stated, viz.,

$$\frac{6FLn}{d} + pS;$$

so that if the effective area of the bolt section be proportioned to sustain this load safely, the error, if any, will be in general on the side of safety.

It is to be remarked that of the two parts of the expression for the whole load on the bolts, the one part, pS, is usually determinable with fair accuracy, whereas the other part, $\frac{6FLn}{d}$, can only be fixed by estimation. In fixing the value to be assigned in any particular case to this latter

part, we may take a step towards a simplification of the expression by assuming that L bears a definite relation to d. For instance, let $L = m \times d$; then the load on the bolts will be

$$6mFn + pS = \text{say } W.$$

Table V has been calculated from this formula, assuming $m = 16$, and $F = 50$ lbs. for a 1-inch bolt and $= 100$ lbs. for a 2-inch bolt, and of proportionate values for intermediate diameters. The figures in the third column represent the maximum test load for good wrought-iron bolts, and are calculated on the basis of a maximum gross stress on the bolt, amounting at the bottom of the thread to about 24,000 lbs. per square inch. The figures in the fourth column represent the test load, if an allowance be made for the unequal distribution of stress among the bolts corresponding to a reduction of 25 per cent. in the effective strength of the joint.

TABLE V.

MAXIMUM LOADING FOR WROUGHT-IRON BOLTS.

Diameter of Bolt.	Stress due to Screwing up $= 6mF$.	Maximum Net Test Load $= \frac{pS}{n}$	Net Test Load $= \frac{pS}{n}$	$\frac{S_1 + S}{S}$
Inches.	lbs.	lbs.	lbs.	
$\frac{3}{4}$	3,600	3,648	2,736	2.31
$\frac{7}{8}$	4,200	5,844	4,383	1.96
1	4,800	8,502	6,376	1.75
$1\frac{1}{8}$	5,400	11,256	8,442	1.64
$1\frac{1}{4}$	6,000	15,582	11,686	1.51
$1\frac{3}{8}$	6,600	20,088	15,066	1.44
$1\frac{1}{2}$	7,200	24,000	18,000	1.4
$1\frac{3}{4}$	8,400	33,360	25,020	1.34
2	9,600	46,080	34,560	1.28

The test loads given in the third column may be adopted when there is a reasonable certainty of the bolts being screwed up so as each to take an equal share of the whole

load; but in general it will be more judicious to limit the test load to the amount given in the fourth column.

Besides being of sufficient strength to resist the maximum load which can be brought on them in ordinary work, the bolts of a joint must also be capable of binding the joint surfaces together with sufficient force to ensure its water-tightness. It is to be observed, however, in this connection, that the water-tightness of a joint does not depend wholly on its forcible compression by means of the bolts and nuts. In the case of a paint joint the adhesion of the paint to the surfaces assists in preventing the passage of water, and in the case of a properly formed guttapercha or leather joint the internal water pressure, acting on the more or less yielding joint packing, assists in rendering the joint water-tight. The initial screwing up of the bolts must, however, put a sufficient pressure on the joint surfaces to bring into play and supplement these assistant actions. It may be taken as a good empirical rule that the pressure on the joint surfaces due to the screwing up of the bolts should be at least equal in intensity per square inch of joint surface to the hydraulic pressure under which the joint is required to be water-tight. This may be expressed symbolically in the form

$$6mFn \nless S_1P$$

if S_1 be the whole area of the joint. Hence there is a certain limiting relation between the area of the joint surface and that of the surface exposed to water pressure for each diameter of bolt. If the limiting relation be exceeded, it will not be practicable for a workman, using an ordinary length of spanner and exerting an ordinary amount of pressure on the end of the spanner, to bring the surfaces together with sufficient force to ensure the water-tightness of the joint. The limiting ratio of S_1 to S is obviously, with the data assumed in Table V., equal to the number in the second column divided by the number in the fourth. The

corresponding ratio of the whole surface to the outside of the joint to the surface exposed to pressure inside the joint or $\frac{S_1 + S}{S}$ is given in column 5 of Table V.

Joints such as those illustrated by Fig. 47 are, however, suitable only for low pressures. For medium and high pressures it is necessary to confine the joint material when used in grooves or recesses, in order that the internal pressure may be prevented from forcing it out, and also to take advantage of the effect of that pressure in adding to the

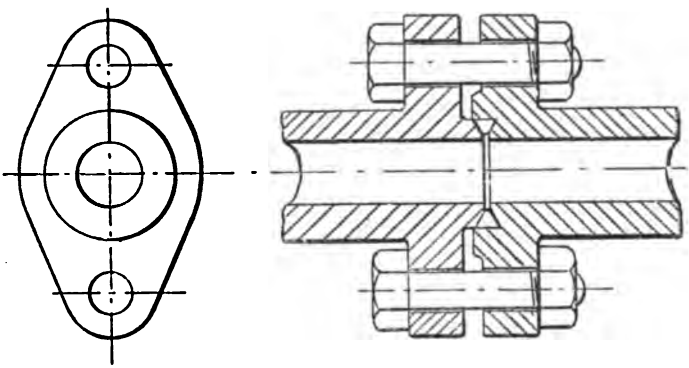


Fig. 48.

water-tightness of the joint in the manner to which we have already alluded. The principles and data which we have exhibited above will still, however, be applicable, as will be readily understood, and may be directly applied to determine the necessary number and dimensions of the bolts.

As a first illustration, we will take the well-known double-lugged Armstrong pipe joint (Fig. 48), so largely used for medium pressures of 500 lbs. to 800 lbs. per square inch.

In this joint a recess about $\frac{1}{2}$ inch wide is turned in the end of one pipe and a corresponding projection on the end

of the next length, which enters the recess, forming a space dovetailed in section, in which a guttapercha ring ¼ inch in diameter is placed. The flanges are drawn together and the guttapercha ring compressed by means of two stout bolts passing through lugs cast on the pipes, as clearly shown in the figure. If D be the inside diameter of the pipe in inches, we have, in this case—

$$S = D^2 \frac{\pi}{4}$$

$$S + S_1 = (D + 1)^2 \frac{\pi}{4}$$

$$\text{Hence} \quad \frac{S + S_1}{S} = \frac{(D + 1)^2}{D^2}$$

Let the test pressure of the pipes, when laid, be taken at 1,600 lbs. per square inch, then—

$$\frac{pS}{n} = \frac{1600 \times D^2 \frac{\pi}{4}}{2} = 628D^2$$

Hence, referring to Table V., for values of $\frac{pS}{n}$, we find ¾-inch bolts will suffice for pipes not exceeding $\sqrt{\frac{2736}{628}}$ in. diameter = 2.09 inches. The corresponding value of $\frac{S_1 + S}{S} = \frac{(2.09 + 1)^2}{(2.09)^2} = 2.19$. Hence the tabular value, viz., 2.31, is not exceeded, and there should be no difficulty in making the joint by an ordinary amount of screwing up. Similarly, 2-inch bolts will suffice for pipes not exceeding $\sqrt{\frac{34560}{628}}$ in. diameter, or 7.42 inches. For pipes of this diameter $\frac{S_1 + S}{S} = \left(\frac{8.42}{7.42}\right)^2 = 1.29$, and the tabular value, viz., 1.28, is slightly exceeded, a result which may be taken as indicating 7 or 8 inches as about the limit beyond which

it is not desirable to employ so small a number as two bolts to make the joint. Proceeding as above, we find—

¾-in.	bolts suitable	for pipes	not exceeding	2.09	in. diameter.
⅞	,,	,,	,,	2.64	,,
1	,,	,,	,,	3.19	,,
1⅛	,,	,,	,,	3.67	,,
1¼	,,	,,	,,	4.31	,,
1⅜	,,	,,	,,	4.9	,,
1½	,,	,,	,,	5.35	,,
1¾	,,	,,	,,	6.31	,,
2	,,	,,	,,	7.42	,,

The lugs may be made 1¾ times the diameter of the bolt in thickness, or a little more—a usual practice in the case of 5-inch pipes, for instance, being to make the pipes 1 inch thick in the barrel, the bolts 1½ inches diameter, and the lugs 2¾ inches thick. The test pressure for such pipes before being laid is usually 2,500 lbs. per square inch, or somewhat in excess of that given in Table II., in which the test pressure for a pipe 5 inches diameter and 1 inch thick is given as 1 ton per square inch.

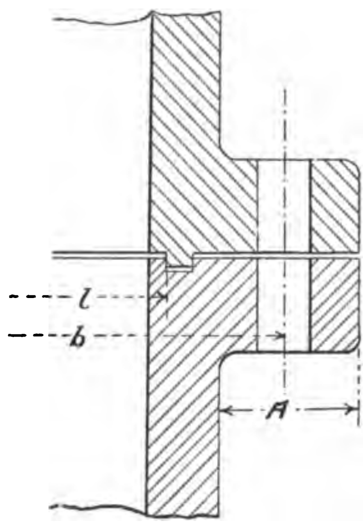


Fig. 49.

Fig. 49 illustrates a form of joint similar to the Armstrong pipe joint, but in which a flat strip of guttapercha is employed instead of a round one as a jointing material, or in place of guttapercha a leather annulus may be used. The

packing ring is here completely enclosed in a recess, and the joint may be used for the highest attainable pressures. The width of the groove need not exceed $\frac{3}{8}$ inch in any case, and may, where desirable, be even less; and its depth may be $\frac{1}{4}$ inch.

If D be the diameter of the pipe, the outer diameter of the groove may be $D + 1\frac{1}{4}$ inches, and its inner diameter $D + \frac{1}{2}$ inch.

$$S \text{ will then be} = (D + \tfrac{1}{2})^2 \frac{\pi}{4}, \text{ and}$$

$$S + S_1 \text{ will be} = (D + 1\tfrac{1}{4})^2 \frac{\pi}{4}$$

$$\text{Hence } \frac{S + S_1}{S} = \left(\frac{D + 1\frac{1}{4}}{D + \frac{1}{2}}\right)^2 = \left(\frac{4\,D + 5}{4\,D + 2}\right)^2,$$

and the test stress on each bolt is—

$$\frac{(2\,D + 1)^2}{n} \times .1963\,p,$$

n being the number of bolts, and p the test pressure in pounds per square inch, as above.

In the case of this description of joint it will not be profitable to employ large diameters of bolts for pipes of small diameters, for the reason that if a sufficient number of bolts be employed to enable a workman with an ordinary length of spanner and ordinary exertion to screw up the joint sufficiently tight to prevent leakage, large bolts, in the case of small pipes, will have an excess of strength to resist the additional stress brought on them by the water pressure. The limiting diameter of pipe for which a particular size of bolt is suitable will be found by equating

$$\frac{S + S^1}{S} = \left(\frac{4\,D + 5}{4\,D + 2}\right)^2$$

for this particular form of joint to the corresponding value

in column 5 of Table V., and hence determining D. Proceeding in this manner, we find that—

$\frac{7}{8}$-in.	bolts	should not be used	for pipes of less than	1.375	in. diam.
1	,,	,,	,,	1.8	,,
$1\frac{1}{8}$	,,	,,	,,	2.17	,,
$1\frac{1}{4}$	,,	,,	,,	2.78	,,
$1\frac{3}{8}$	,,	,,	,,	3.25	,,
$1\frac{1}{2}$	,,	,,	,,	3.6	,,
$1\frac{3}{4}$	,,	,,	,,	4.25	,,
2	,,	,,	,,	5.22	,,

The thickness of the flanges of pipes of the type illustrated by Fig. 49 is more properly a function of the diameter and pitch of the bolts than of the thickness of the pipe metal. The following is a rule which may be used with advantage to determine the proper thickness of the flange. Let d_8 be the diameter of the bolts in eighths of an inch, and c the pitch or distance between the centres of two adjacent bolts, measured along the arc of the circle of bolt centres in inches, then the thickness of the flange should not be less than

$$\frac{d_8}{9}\sqrt{\left(\frac{d_8+3}{c}\right)}$$

the width of the flange (dimension A, in Fig. 49) being equal to

$$\frac{d_8+3}{4}$$

Hence we obtain the dimensions tabulated in Table VI. for joints of the type illustrated by Fig. 49.

If the thickness of flange, as given by the table, for any particular case be less than the thickness of the barrel or body of the pipes, the thickness of the flange should be made greater than that given in the table, or say not less than the thickness of the pipe.

Table VI.

Dimensions of Circular Flanges of Cast-Iron Pipes with Tongued and Grooved Joints.

Diameter of Bolts.	Width of Flange A.	Thickness of Flange in Inches. Pitch of Bolts in Inches.							
		3	4	5	6	7	8	9	12
In.	In.	In.	In.	In.	In.	In.	In.	In.	In.
$\frac{3}{4}$	$2\frac{1}{4}$	1.16	1	.9	.82	.76	...	...	...
$\frac{7}{8}$	$2\frac{1}{2}$	1.43	1.23	1.11	1.01	.94	.88	...	...
1	$2\frac{3}{4}$	1.7	1.48	1.33	1.21	1.12	1.05	1	...
$1\frac{1}{8}$	3	2	1.73	1.55	1.42	1.31	1.23	1.16	1
$1\frac{1}{4}$	$3\frac{1}{4}$	...	2.01	1.8	1.65	1.52	1.42	1.35	1.16
$1\frac{3}{8}$	$3\frac{1}{2}$	...	2.3	2.05	1.88	1.73	1.63	1.53	1.33
$1\frac{1}{2}$	$3\frac{3}{4}$	...	...	2.33	2.13	1.97	1.84	1.73	1.5
$1\frac{3}{4}$	$4\frac{1}{4}$	...	...	2.88	2.63	2.43	2.27	2.15	1.86
2	$4\frac{3}{4}$	...	...	...	3.17	2.94	2.74	2.59	2.24

The strength of flat cover plates is open to considerable doubt, but the following formulæ give results which are found efficient in practice—

$$\frac{bt^2}{6}f = Wl \times .1,$$

in which b is the distance between the centres of bolt holes in inches, t thickness of plate in inches, f the stress in tons per square inch to which the metal is to be stressed, W is the total load on the cover in tons, and l the inside diameter of the spigot.

According to the above rule, the cover for a 6-inch cylinder of a hotel lift working at 750 lbs. per square inch requires to be about 2 inches thick—

$b = 10.5''$, $f = .9$ ton (from Table I.), $W = 12.6$ tons, $l = 7''$.

$$\frac{10.5 \times t^2}{6} \times .9 = 12.6 \times 7 \times .1.$$

$$t^2 = 5.5$$
$$t = 2.3''.$$

By Grashof's rule—

$$t^2 = \frac{2}{3}\frac{r^2 p}{f},$$

where r is the radius in inches, and p the pressure per square inch in tons, so that

$$t^2 = \frac{2}{3} \times \frac{12.2 \times .3}{.9}$$

$$t = \sqrt{2.7} = 1.7''.$$

Flat cylinder ends are only suitable for very small sizes and low pressures, owing to their great thickness for moderate strength. For large cylinder covers the dished form is generally employed and shown in Fig. 16 (*ante*) and Fig. 50. When there is a joint, as in Fig. 50, the rise V should be about one quarter the diameter, and the thickness of the cover the same as the sides of a cylinder of diameter l. The cover and cylinder will then have about equal strength. The question of bolts has been already dealt with.

Fig. 51 illustrates the old method of joint for a long hydraulic main, while the more modern method adopted by the London Hydraulic Power Company is shown in side elevation and section in Fig. 52. The joints are of the spigot and faucet type, turned up with a V groove, in which is inserted an indiarubber or guttapercha ring. The pipes are made in about 9-foot lengths, and are held together by the bolts passing through the lugs at the end of each length. In the old form of pipes the face of the lugs was nearly flush with the end of the pipe; but in this new form shown in Fig. 52 the lugs are set back some distance from the end,

an improvement which has been found to increase the strength some 35 per cent., very few failures of lugs having occurred since this form was introduced by the Company,

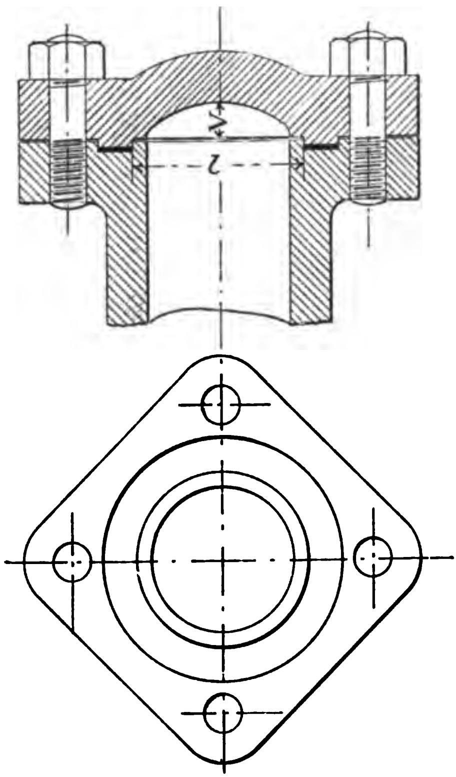


Fig. 50.

whereas with the old type of lugs failures were not uncommon. Fig. 53 is a full size section of the rubber ring when compressed in the V groove.

Fig. 54 illustrates the ordinary socket and spigot joint

used in long mains, in which the pressure does not exceed 250 lbs. per square inch. After placing the spigot end of one length in the socket end of another, and ramming into

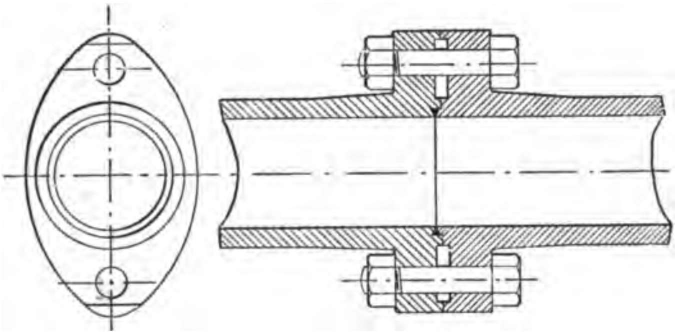


Fig. 51.

the bottom of the socket some greased hemp, the joint is made by pouring in molten lead. The lead by running into the groove A round the inside of the socket prevents the pressure from forcing the plug of lead out. If the main

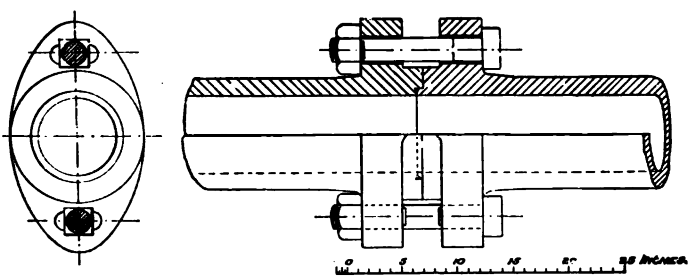


Fig. 52.

is intended for a permanency, the socket may be filled with a rust joint cement in place of lead. A good joint composition is as follows:—2 parts by weight of sal-ammoniac,

1 part flour of sulphur, 200 parts iron borings; the whole made to a paste with water. This mixture makes a lasting cement, although a slowly setting one, and is one not to be used when the pipe is required for immediate service.

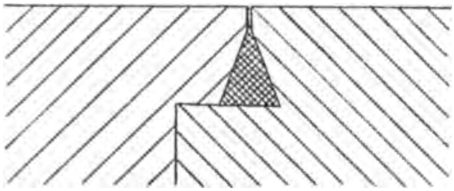


Fig. 53.

The drawback to a rust-joint is that the pipes must be broken if any alteration to the main is required, as the cement sets harder than cast iron, if properly made, whereas with a joint made with lead the lead can be cut out if the joint is to be broken. In socket and spigot jointed mains

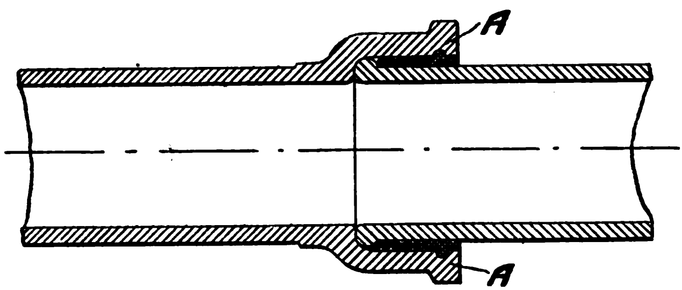


Fig. 54.

it is a good practice to put flange joints every 100 or 150 feet run for the convenience of alterations or repairs.

When a pipe main is laid on the surface of the ground, exposed to the varying temperature between day and night, expansion joints (Fig. 55) are sometimes put in the main

at intervals of 400 to 500 feet to obviate the tendency to crack, and to prevent the creeping of the joints, which commonly causes leaks.

The expansion joint shown in Fig. 55 is formed by turning the spigot end of one length of pipe to work through a bored gland and stuffing box cast on the socket end of another length of pipe. The gland and stuffing box are bushed with gun metal, and the gland packed with hemp in the usual way. In an exposed main it is necessary to anchor the stuffing box length of pipe firmly to a concrete or stone block to prevent its tendency to creep. Especially is this necessary if the main is on an incline instead of lying horizontally, for gravity will then assist the creep of the pipe down the incline.

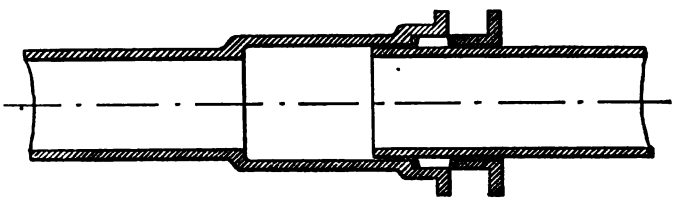


Fig. 55.

An exposed main of cast-iron piping, some 500 feet long, will vary on the average 1 inch in its length between midday and midnight in the summer season; but this amount of expansion will be reduced to about .3 inch if a stream of cold water be kept rapidly and continuously running through the pipe.

It is not always possible or convenient to arrange cast-iron mains or conduits for conveying the hydraulic pressure, in which case it is desirable to be able to attach, at any required position upon the pipe employed, a means of connecting one portion with another, or of attaching a branch to the main supply.

Pipes of wrought iron, steel, or copper, under 3 inches diameter, may be very readily jointed together for low pressure by means of a right and left hand screw coupling socket nut, which draws the ends together into metallic contact;

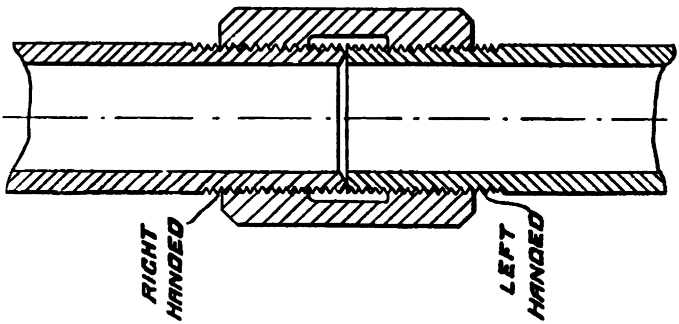


Fig. 56.

the end of one pipe being turned truly flat, and the other to a truly sharp edge, as shown in Fig. 56. The objection to this mode of coupling arises from the difficulty experienced

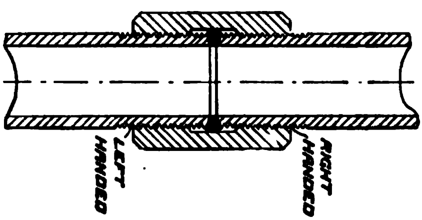


Fig. 57.

in releasing the pipes, it being impossible to undo the joints unless the pipes have room to separate when the nut is unscrewed, which, in many cases, would be quite impracticable. A similar mode of jointing is shown in Fig. 57, in which a

rubber ring is inserted to make the joint, but of course the same objection applies in this case as to the former joint.

The more common, although more costly, method of jointing pipes is illustrated at Fig. 58.

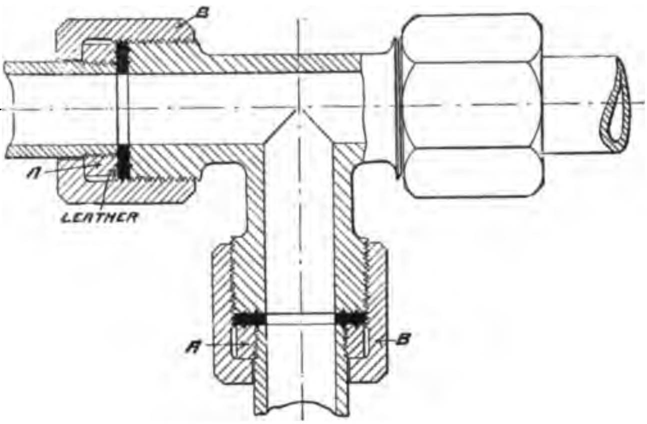


Fig. 58.

The end of one pipe is screwed to receive a collar A, and before this collar is placed upon the screwed portion a nut B is passed over the pipe, so that the nut is then made, as it were, a part of the pipe. The end of the junction piece, or T-piece, is also similarly screwed, and a leather washer is inserted between the ends, as shown. The connection of copper pipes is usually effected by the method illustrated at Fig. 59, the socket being brazed on to one and the flange brazed on to the other end, having first been screwed on their respective pipes.

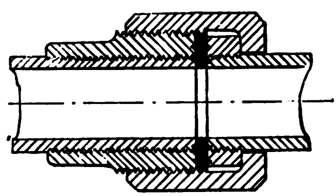


Fig. 59.

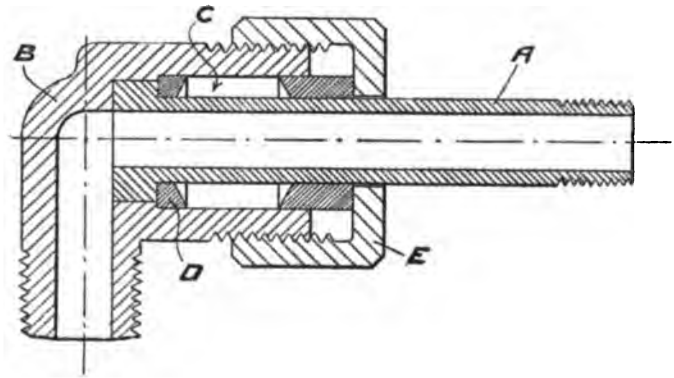


Fig. 60.

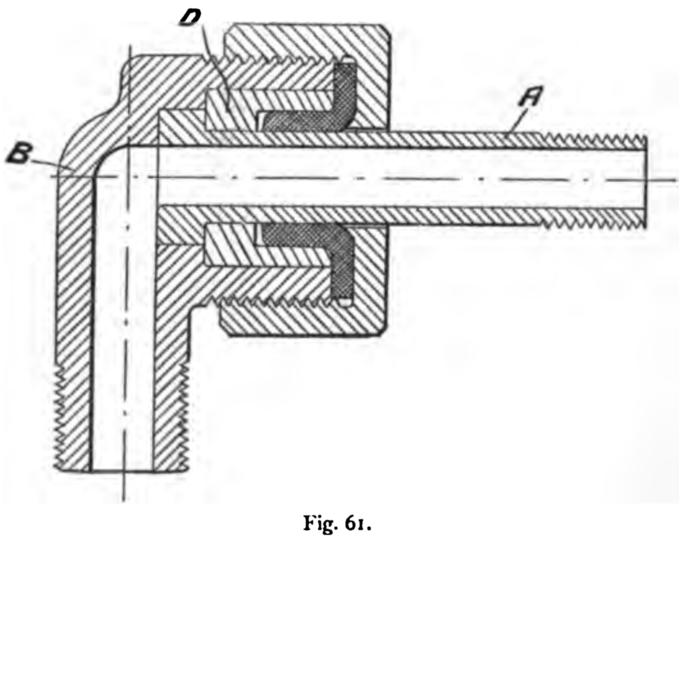


Fig. 61.

With the application of hydraulic power to cranes, riveting machinery, etc., swivelling or turning joints for the walking pipes are a necessity. Fig. 60 illustrates a gun-

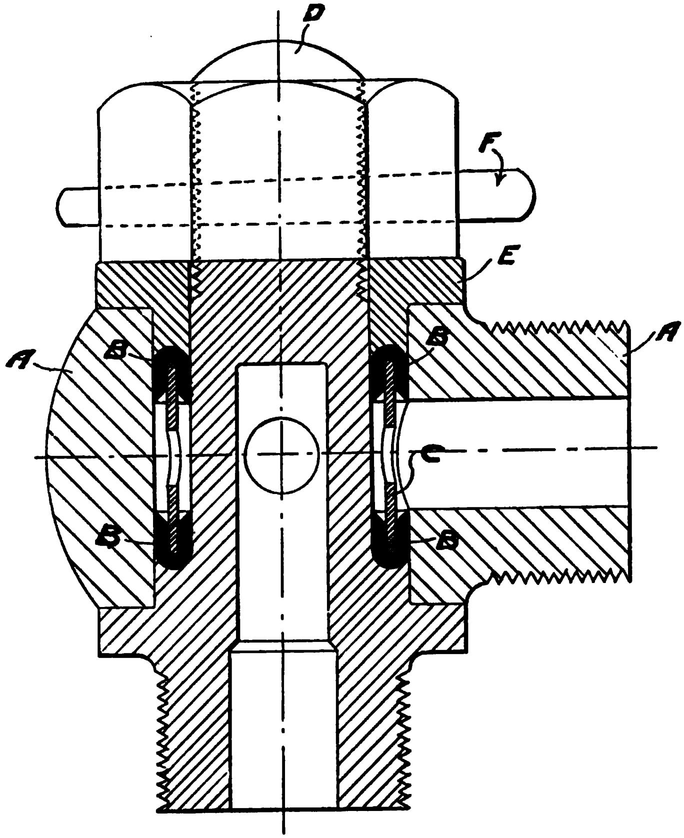


Fig. 62.

metal right-angle swivelling connection for a pressure of not more than 700 or 800 lbs. per square inch. It consists of a flanged pipe A turning easily in the elbow piece B, having

the stuffing box C enlarged so that the ring D may seat on the shoulder and relieve the flange of the pipe A from any pressure consequent upon screwing down the gland E. Fig. 61 shows the same kind of swivelling connection, but having a hat leather packing in place of a stuffing box. Both these

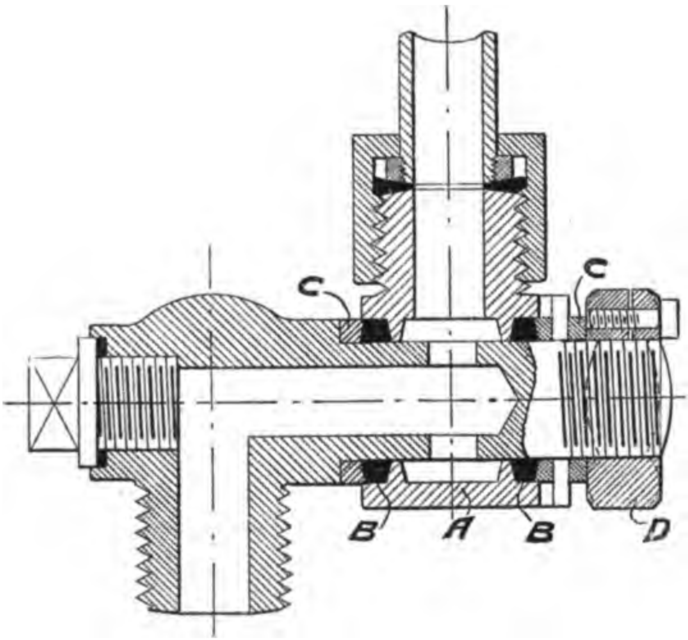


Fig. 63.

types answer well, but have the one drawback of the pressure acting on the sectional area of the pipe thickness and forcing the flange of the pipe A against the ring D to an extent which prevents this form of connection being used for higher pressures than above stated. To obviate this the joint shown in Fig. 62 is adapted, in which the swivelling

piece A is packed by two U leathers B, which are kept apart by the brass ring C, this ring being drilled with holes for the passage of the water. The leathers are secured in their position on the pin D by means of the washer E and nut and cotter F. If due care is taken in its manufacture, this joint is thoroughly reliable, with pressures up to 1,600 lbs. per square inch, and lasts a long time before requiring renewal of the packing. Fig. 63 illustrates a similar connection, but with plain leather washers for packing in place of the U

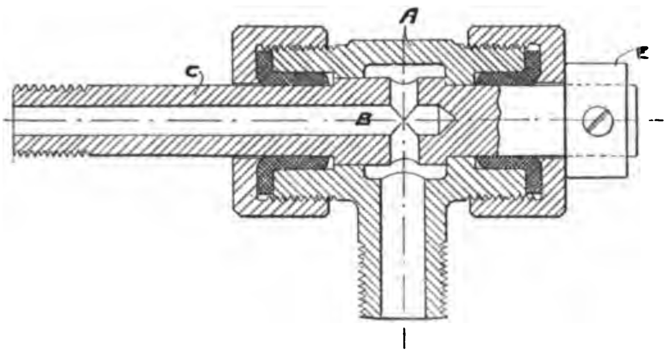


Fig. 64.

leathers, as shown in Fig. 62. The swivelling piece A has a shallow stuffing box B at each end, for which the rings C C act as the glands, these glands being fitted with pegs so as to turn with the piece A, and they can be tightened up by means of the locking nut D.

Fig. 64 shows a swivelling joint suitable for a pressure of 3 to 4 tons per square inch, in which hat leather packings are employed. The hollow pin or pipe C has an enlarged end at B round which the joint A revolves, and is secured from sliding endways by the set collar E. Sometimes the

hollow pin or pipe C has the swell B made the whole width of the turning joint A, in which case two set collars are required, one at each side of the turning joint, and close to the gland nuts, to retain the joint A in position. This last arrangement has the advantage that it permits of the introduction of fresh leathers without disconnecting the pipe C.

PART IV.—VALVES.

CHAPTER VIII.

CONTROLLING VALVES.

Of all the auxiliary mechanism employed in hydraulic power works the valves are the most important, for on their efficient working depends the success of the undertaking.

The design of valves for hydraulic machinery varies according to the purposes for which that machinery is intended, and the constant applications for patents in connection with hydraulic valves must be taken as evidence of the importance of the subject, and at the same time as a proof of the necessity for the special attention which is necessary in designing any hydraulic valve.

In the present chapter it is intended to point out some of the leading features that go to make a successful working valve, and then to describe in detail some of the more common types of valves.

Fig. 65 illustrates an ordinary form of stop valve for medium pressures consisting of a cast-iron body A, having lugs for connecting to the pressure pipes forming the hydraulic main, and provided with a cap secured to the valve body by the studs B. A hard gun-metal valve seat is screwed into the body at C, making a tight joint by means of the rubber ring. The cap has a tapped gun-metal bush D, in which works the screwed stalk of the gun-metal valve spindle E; the bottom of the stuffing box has a gun-metal bush F, and a gland ring G presses upon the packing when the cap is screwed down.

If H is the diameter of the bore in the bush, the valve seat of which is angled off at 45°, and the end of the valve spindle is level with the bottom of the mitre seat

when the valve is shut, then the required lift of the valve spindle E off its seat so as to have an annular space between it and its seat equal in area to the water passage H is .305H; but in order to lessen the loss of head consequent upon the

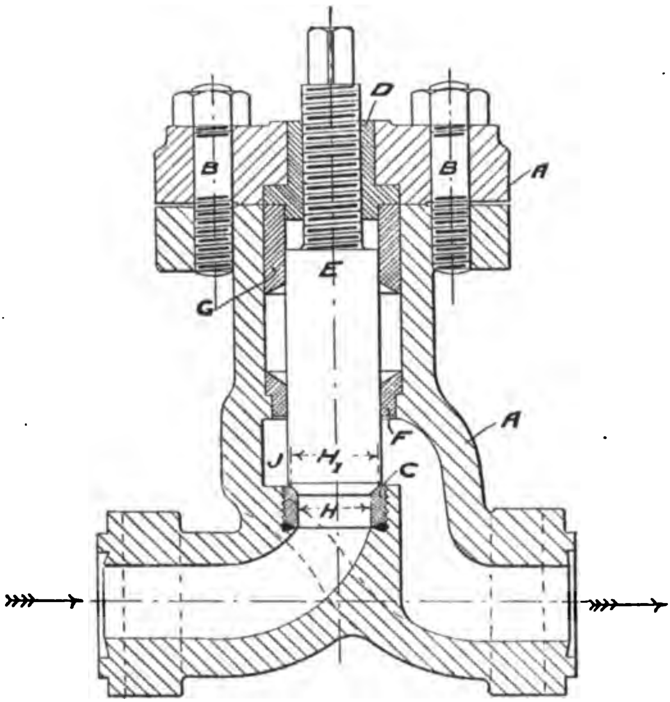


Fig. 65.

flow of water through the valve the lift of the spindle E is made from .375H in large valves to .5H in small ones.

For a similar reason the sectional area of the annular space J round the spindle should not be less in width than .375H.

In general practice it is better to shut the valve against

the flow of water than with it, for the reason that the water pressure on the spindle causes all backlash in the screw-threads and other parts to be taken up before the closing of the valve.

To prevent leakage, the pressure of the spindle E upon its mitre seat C per square inch of seat surface requires to be at least equal to the water pressure per square inch. Let H and H_1 equal respectively the inner and outer diameter of the mitre turned on the valve seat, also let p be the water pressure per square inch and P the least total pressure on the valve spindle E to ensure the water not leaking through, then

$$P = \frac{H_1^2}{4} \times p\pi = H_1^2 \times .7854p.$$

We may now determine the size of a hand-wheel for, say, a $1\frac{1}{2}$-inch stop valve for 750 lbs. pressure per square inch. Let x equal the diameter of hand-wheel, and assume a man can exert a maximum turning effort of 120 lbs. on the rim of the hand-wheel. For a valve of this size the spindle E would be about $1\frac{7}{8}$ inches diameter, and the pitch of the $1\frac{1}{4}$-inch screw cut upon the stalk about 6 threads per inch. In this example there are four resistances to be overcome by the hand-wheel, viz., P, the pressure; the friction of the valve when turning on its seat at the instant of closing, which, taking .3 as the coefficient of friction, equals $.3p(H_1^2 - H^2)\frac{\pi}{4}$; the friction of the spindle in its stuffing box, which may be obtained from Table IV., thus $(1 - .93)P$; also the friction of the screw due to the pressure P, the coefficient of friction being in this case .15.

For one revolution of the hand-wheel the work done amounts to $120 \times x \times \pi$, which must balance the resistances:

$$\left.\begin{array}{l}(1.)\ P \times \frac{1}{6}'' + \\ (2.)\ .3p(H_1^2 - H^2) \times \frac{\pi}{4} \times 1\frac{3}{4}'' \times \pi + \\ (3.)\ (1 - .93)P \times 1\frac{7}{8} \times \pi + \\ (4.)\ .15P \times 1'' \times \pi\end{array}\right\} = 120 \times x \times \pi.$$

Solving this equation for x we get the above example, 8.5 inches as the diameter of the hand-wheel.

In large stop valves, from about 4 inches and upwards, it is found necessary to attach a balancing arrangement, otherwise one man would not be able to open or close them.

Fig. 66 illustrates a similar stop valve to that shown by Fig. 65, but having its valve spindle packed by a leather lace

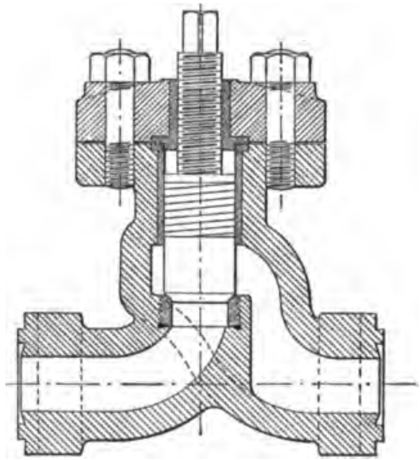


Fig. 66.

instead of the ordinary stuffing box. This method of packing answers very well for valve spindles not more than $1\frac{1}{2}$ inches diameter, but for diameters above $1\frac{1}{2}$ inches the stuffing box form of packing should be adopted.

Where a number of hydraulic tools are at work it is advisable to put in the main a safety valve, for the simultaneous stopping of several tools will so suddenly check the falling accumulator as to augment the normal pressure to a dangerous extent unless it can find relief. The safety or

shock valve shown in Fig. 67 is designed for this purpose, and consists of an ordinary cast-iron T-piece, having flanges for bolting to the pipes forming the hydraulic main, the stalk of the tee piece being provided with a gun-metal mitre valve and seat, while the valve is loaded by a combined adjustable spring and dead-weight lever. The minimum pressure is put on by the spring by adjusting the height of the cross-head and locking the nuts, and the additional pressure above

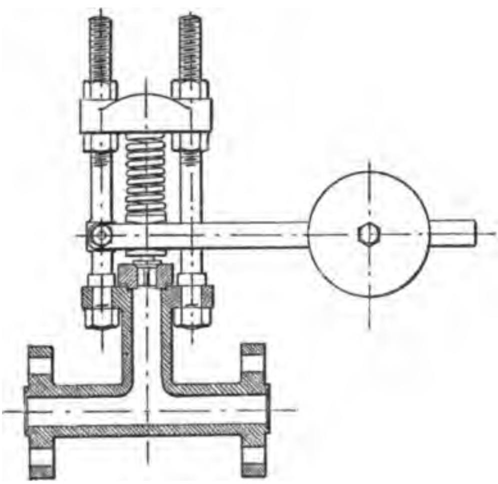


Fig. 67.

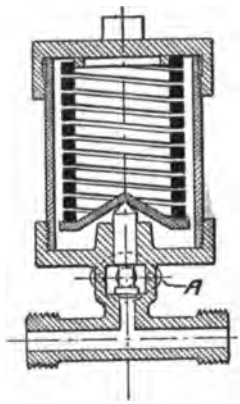


Fig. 68.

that of the accumulator is obtained by adjusting the position of the weight upon the lever.

Fig. 68 illustrates a closed-up spring-loaded safety valve, of which the body is made entirely of gun metal with an overflow pipe at A. The point of suspension of the spring-loaded plate is above the plane upon which the spring bears to ensure stable equilibrium. This form of relief valve prevents any tampering with it after the spring is set to allow the valve to lift at a given pressure.

Although safety valves relieve the pipe of stress from excess of pressure, they have the disadvantage of allowing the water that flows through the valve to run to waste. To obviate this the arrangement as illustrated by Fig. 69 is employed, which is called a shock or relief valve, and consists of a closed-up spring-loaded small ram working through a stuffing box and gland in a cylinder having branches for connecting to the pipes of the hydraulic main. The ram is loaded by the spring to the working pressure by the method shown in Fig. 68, and when the pressure through any cause rises above the normal the ram is raised, and thus the pipe is relieved of any excessive stress that would occur if there were no relief. The spring can be either cylindrical or of volute form, but in any case it must be sufficiently long to admit of a large deflection without much increase of pressure. The apparatus is practically a small accumulator.

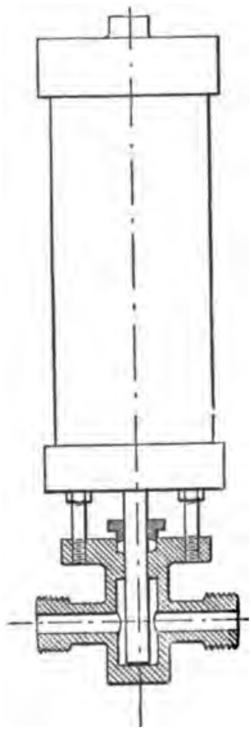


Fig. 69.

The London Hydraulic Power Company place a shock valve on each side of every stop valve in their 6-inch pressure main, and in most hydraulic plants worked by an accumulator it is advisable to put a shock valve in the delivery main close to the accumulator.

In most forms of hydraulic machinery worked by pressure energy that part of the mechanism which is acted upon directly by the water pressure consists in some form or other of a ram working in a cylinder rendered water-tight by means

of a hemp or leather packing, such as the ram of a press or lift, and the function of the valve is to admit the water from the pressure pipe to the cylinder, and then to close the admission when the ram has run out sufficiently far, and finally to open the cylinder to exhaust so that the water within the cylinder may run to waste while the ram is returning in

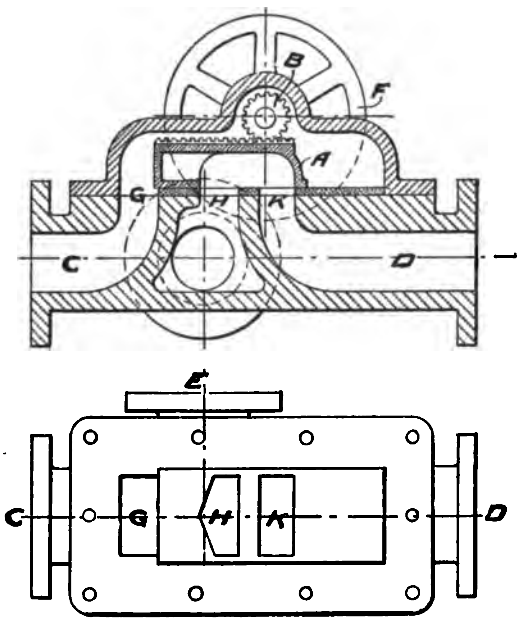


Fig. 70.

most cases without hydraulic aid. The type of valve in common use for low-pressure lifts is shown in Fig. 70, and is termed a rack slide valve. A is the gun-metal valve sliding on a gun-metal face pinned to the cast-iron valve body. The valve is worked by a rack on its upper side engaging a pinion B, which is fast on the axle of the rope wheel F. An

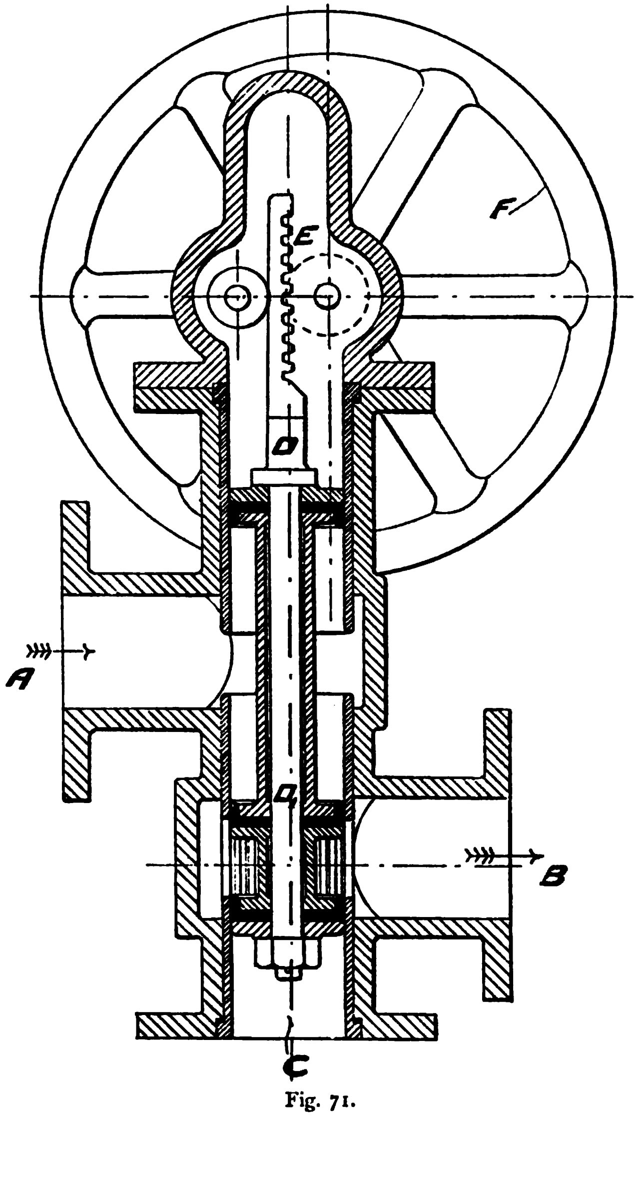


Fig. 71.

endless rope engages this wheel, one end of which passes up through the cage or platform of the lift. C is the pressure inlet, E the branch for connection to the lift cylinder, D the outlet or exhaust, G the pressure port, always open, H the port leading to the cylinder, and K the exhaust port. The side of the port H opening to pressure is often cut in the shape of a large **V**, so that the closing of this port to pressure may be effected more gradually and thereby reduce the chance of any shock. The valve is shown in the position when the cylinder is fully open to exhaust, and on pulling the rope so as to move the valve A to the right, the exhaust is closed, in which position the valve face should lap at least $\frac{1}{4}$ inch over each side of the port H to ensure no leaking. Upon moving the valve further to the right it uncovers the port H to pressure. This form of valve is particularly convenient for any kind of hydraulic lift or crane, as the rope working the valve can be led away in any direction.

Sometimes small shock valves are inserted in the slide valve body, but for low pressures the general practice is to put an air vessel between the valve and the cylinder to reduce the effect of shock. The rack slide valve is seldom used for larger inlets than $3\frac{1}{2}$ to 4 inches, as the friction of the valve on its face is then more than can conveniently be overcome by one man, and the type of valve shown by Fig. 71 is then generally adopted for low-pressure lifts. It consists of a leather packed gun-metal piston valve D_1, and rod D actuated by a rack and pinion E and working in a gun-metal lined cast-iron valve body, having the branches A to pressure, B to cylinder of the lift, and C to exhaust. The gun-metal liner has narrow vertical slots or holes cut round it, opposite the branch B, and these slots are covered by the piston valve D_1, when in the middle of its stroke. F is the rope wheel round which a cord is wound and led away to the lift. As the pressure is acting on equal piston valve areas the valve is balanced, permitting the wheel F to be easily revolved.

When the piston valve is lowered so as to uncover the top

end of the vertical slots, the pressure passes from A along the branch B to the cylinder, and when the piston valve is raised so as to uncover the bottom end of the slots, the water in the cylinder can pass by the branch B into C, and thus to exhaust.

Slide valves cannot be successfully used for pressures exceeding 1,600 or 1,700 lbs. per square inch, and although many attempts have been made to automatically balance them, failure has invariably been the result, owing to the fact advanced at the beginning of this chapter, viz., that a valve to be tight must be pressed upon its seat with at least an equal pressure per square inch to that of the water.

Fig. 72 illustrates a slide valve similar in working to the one shown in Fig. 70, but modified in design, for pressures up to 1,600 lbs. per square inch. A is the connecting branch to the pressure main or supply, B is the branch to the exhaust, and C is the branch to the cylinder. This valve in the smaller sizes is usually made of gun metal throughout, having a loose face K pinned on to the body.

The valve rod L is enlarged in the middle of its length, and has a hole cut in it to receive the stalk of the valve D. The rod works through packed glands at each end, and is so arranged that it can be withdrawn through the stuffing box. The ports F and G, which lead into the branches B and C, are opened and closed by the slide valve D, and the enlarged part of the rod L prevents the valve moving too far either way.

Should the lift or crane which is worked by this valve be suddenly checked, when lowering with a heavy load, by moving the valve to close the port G, the pressure in the cylinder would be augmented above the working pressure. This excess pressure then finds relief through the bye passage H and small flap valve E. This small valve E then becomes a shock valve, and is usually made in the form of a weighted leather washer, pinned or screwed to the face of the valve body, as shown in the plan. The advantage

in making the ports G circular is the possibility of a more gradual opening and closing of the ports than is obtained with rectangular openings.

There are various ways of operating this valve. For instance, the valve rod L can be connected to a rack and

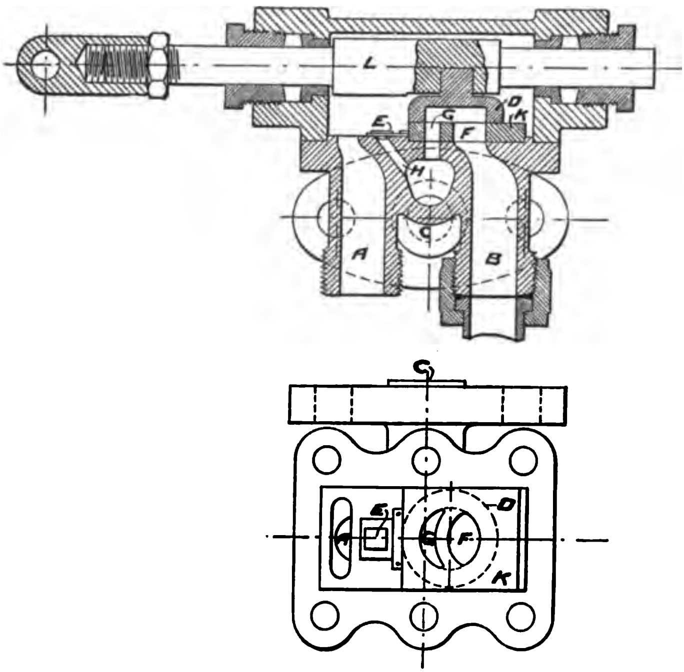


Fig. 72.

worked by a pinion and rope wheel, as in Fig. 70, or it can be readily worked by a combination of levers.

When the pressure exceeds 1,600 lbs. per square inch the valve should be of the design illustrated by Fig. 73, first employed by Lord Armstrong's firm for crane purposes.

The valve can be worked vertically or horizontally as may be desired. A is the pressure inlet, B exhaust outlet, C the passage to the press or lift cylinder; D and E are the valve

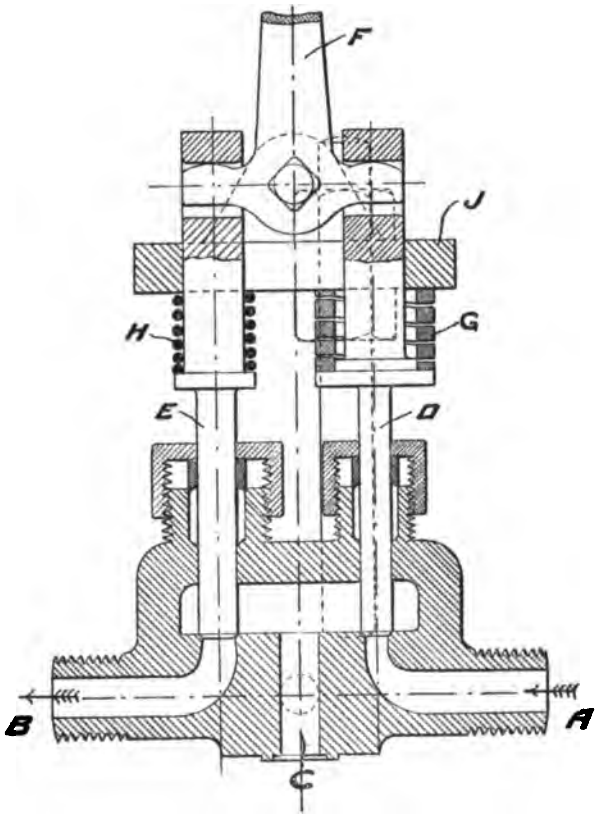


Fig. 73.

spindles working through stuffing boxes, and closing the ports to A and B respectively. These spindles are kept down on their seats by means of the springs G and H bear-

ing against the crossplate J, this latter being secured to the valve body by two bolts. The proportions or sizes of the springs may be determined by the method stated at the beginning of this chapter. The valve spindles are lifted by means of a T-shaped lever F. On pulling the lever to the left or right the spindle valve D or E is raised, and on releas-

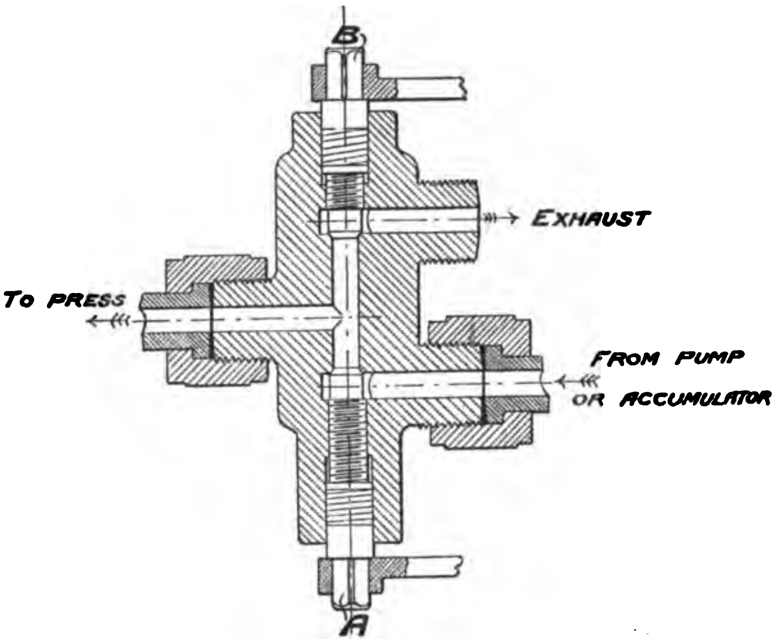
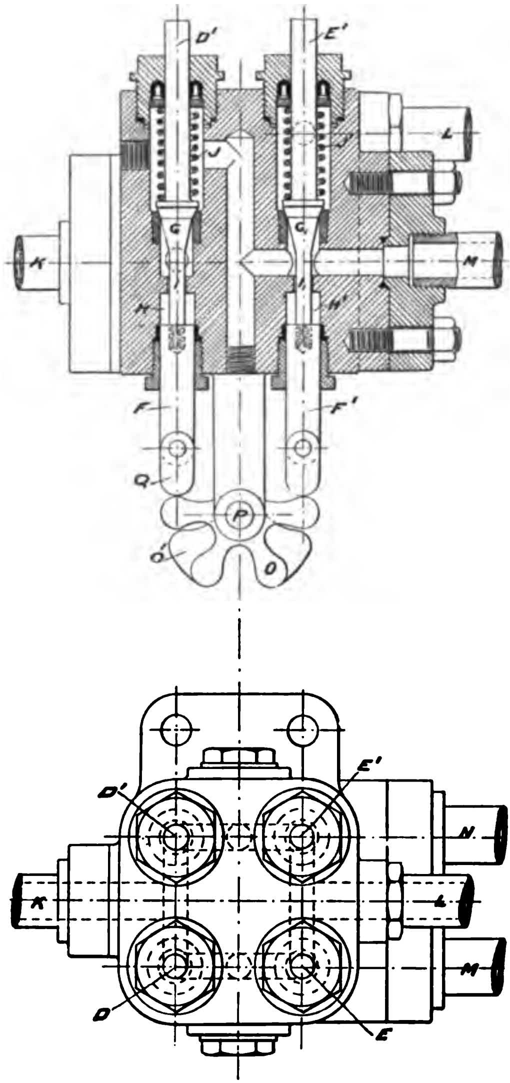


Fig. 74.

ing the lever the valves automatically close the ports. The larger sizes of this design of valve are fitted with a small shock relief, as already described.

For heavy pressures up to 3 and 4 tons per square inch the simplest and most convenient type of valve is illustrated by Fig. 74. It is usually made entirely of gun metal, the



Figs. 75 and 76.

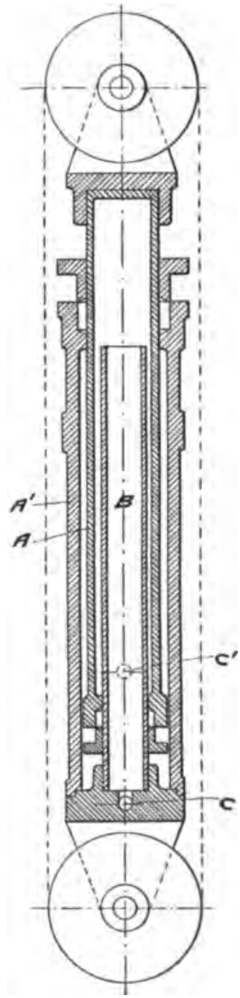
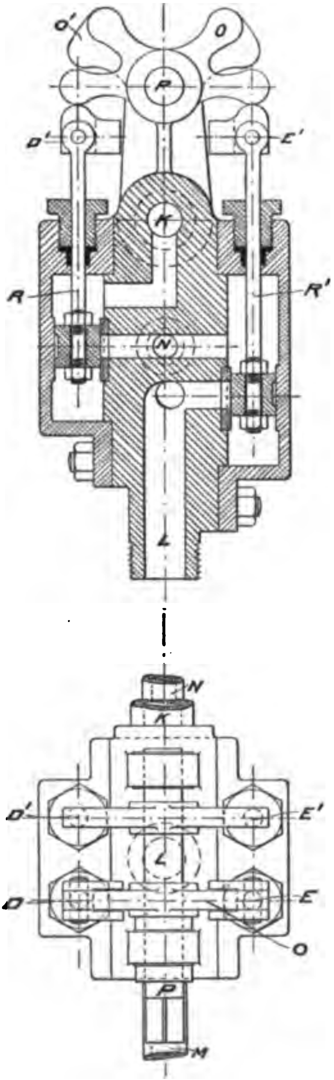


Fig. 77.

valve spindles A and B being packed with leather laces and fitted with handles. In using this valve the spindle A is opened first, admitting pressure to the cylinder of the press; it is then shut, and the valve spindle B opened, allowing the water from the cylinder to exhaust, care being taken not to have both valves open at once, or the pressure water will run to waste. To obviate the possibility of both valves being opened at one time the author has designed valves with spindles placed side by side, and actuated by means of gearing working right and left hand screwed valve stems.

A good example of a partially balanced spindle valve is shown in Figs. 75 and 76. Fig. 75 is a sectional elevation, and Fig. 76 the plan of Meacock's valve for admitting the pressure to the cylinders of the two power jigger or multiple chain lift shown in section by Fig. 77, in which C is the inlet to the cylinder containing the small ram, and C^1 the inlet to the cylinder containing the larger ram. The valve arrangement consists of four plug valves D D^1, E E^1, which are held upon their respective seats by the water pressure acting upon the increased area of their spindles F F^1 over the areas of the valve ports G G^1. The chambers H H^1 above the spindles F F^1 are charged with water. This water has to be displaced during the rising of the plug valves, and they can only automatically return upon their seats as the chambers H H^1 become charged with water through the clearance effected by the diameter of the valve stalks I I^1 being less than the diameter of the passages in which they work, thereby ensuring steady action. The springs J J^1 are for the purpose of keeping the plug valves D D^1, E E^1 upon their seats when water pressure is shut off from the supply main. The pressure inlet is marked K, and is common to both of the valve plugs D D^1. The exhaust outlet is marked L, and serves for both the valve plugs E E^1. The pipe M communicates with the internal ram B through the inlet C, and the pipe N is connected at C^1 to the cylinder A^1 containing the ram A. The valve plugs D D^1, E E^1 are



Figs. 78 and 79.

actuated by the two double cams O O^1 fixed on the spindle P. By partially rotating the spindle P in one direction by means of a wheel or a lever fixed to it, the valve plug D is raised, thereby admitting pressure to the ram B, and by further rotating the spindle P the valve plug D is liberated by means of the slipping link Q, when it automatically seats itself, thereby closing communication to the ram B. During this time the cam raises the valve plug D^1, so that the annular area of the ram A is admitted to pressure. For raising a maximum load a third movement of the cam again raises the valve plug D, at the same time retaining the valve plug D^1 open, both rams now being subjected to pressure.

If the spindle P is turned in the opposite direction, thus causing the cams to operate upon the exhaust valve plugs E E^1, either the small cylinder B or large cylinder A^1 may be opened to exhaust, or by a further movement both may be opened. By this means a great economy of pressure water is effected.

An arrangement of four slide valves may be used in place of the four spindle valves just described, as shown in sectional elevation and plan in Figs. 78 and 79. The slides are made to automatically cover the ports leading to the cylinders A^1 B by the pressure acting upon the valve spindles D D^1, E E^1, as shown at R R^1. These slides are caused to open the ports to admit pressure to the cylinders A^1 B, or to put them to exhaust by the action of the pair of double cams O O^1 fixed upon the spindle P, and operating in a similar manner to that described for the opening of the plug valves. The inlet K is connected to the pressure main, and the outlet L to the exhaust, while the pipe M communicates with the inlet C, and the pipe N with the inlet C^1.

Brindley's patent water pressure balanced pilot valve controlling a larger main valve is employed with advantage when a small movement is desirable for the operating lever as in connection with riveting plants and hydraulic presses.

PART V.—LIFTING MACHINERY.

CHAPTER IX.

PLATFORM LIFTS.

One of the most popular applications of hydraulic power is connected with lifting machinery, when passengers or goods are raised from floor to floor of lofty warehouses, or for general manufacturing premises. The question of correct working is greatly misunderstood, and what is far more serious the safety of such lifts is only too often a matter quite ignored by those responsible for the working of the machines. It is said that any person can construct a lift, for the pressure is on the water, and the only thing remaining for the constructor is to make a simple machine to transform this pressure into mechanical power. Then again, too, safety appliances are mentioned as being specially provided to meet any emergency which is likely to arise, so that the possibility of danger of accidental occurrence is a matter to be treated with equanimity by those about to trust their lives in such machines; whereas the fact is only too painfully advertised that but few persons can properly construct and erect a lift which is at once economical, safe, and simple in principle.

There is probably no piece of machinery subject to more unfair usage and more rough and careless handling than the hydraulic lift, for it is to be everybody's assistant, and every one handles it in a manner that he or she considers to be the best way. We have known valves to be pulled violently backwards and forwards by warehouse and factory lads and girls, causing shocks and strains to be given to all parts of the machinery, which have produced permanent injury and sometimes disaster; while in many cases fatal accidents, attributed to the lift, and reported as "another

lift accident" in the daily journals, may be clearly traced to reckless and contributory negligence on the part of those injured. Similarly, the so-called safety appliances seldom prove of service in the cheap and common lift, for being always in a stationary or fixed position during the normal working, they get quite stiff, rusty, and clogged up with dirt and grease, and refuse to act when suddenly they are liberated after long standing unused.

To be of any practical or real service as safeguards, the appliances which are supposed to arrest the motion of the cage or lift platform when an accident occurs, such as the severing of a cable or chain, or the disconnection of a ram, should always be in actual use or work. They should form the absolute and definite base upon which the motion of the car or platform depends, so that in the event of any failure occurring the gear at once comes into play, and does its part promptly and well. When this condition of construction is more fully understood, we shall hear less of such accidents, which have made lift-users tremble in the past, and which have caused the demands to be made for compulsory registration of all passenger hoists and lifts. The author considers that every lift should be under the supervision of the Board of Trade, and licensed before being allowed to carry passengers.

There is in many minds a strong prejudice against being *pulled* up by any mechanical appliance used in connection with hoists and lifts, while the same feeling does not appear to be induced when the persons are *pushed* up. Thus it is that nervous persons entering a lift, which is suspended by chains or ropes, sometimes reflect as to what will happen to them in the event of such chains or ropes giving way or failing. They do not allow any feeling or question of failure to trouble them when they are unable to see the mechanism which operates the lift; they simply conclude that it is something they cannot understand, because it is not immediately before their eyes. To this class of person a ram lift

is quite safe, and greatly to be preferred to any suspended type; whereas the fact remains on record that the most serious accident which has happened to any public lift occurred upon a direct-acting or ram lift. There are elements of danger everywhere, but probably the safest place in the world, taking the number of persons carried into account, and the careless handling that controls the working of lifts generally, is a car of a modern high-class suspended elevator.

A good lift provides for every contingency which can befall it: excessive speed, overloading, failure of the valve, breakage of the ram or suspending cables—all of these are properly anticipated by the high-class maker; but, as in the case of every refinement, they have to be paid for in the first instance. Here it is that cheap and common lifts come in and secure a market; they are capable of raising as much load, and at as quick a speed, as the good and safe lift, while they cost about 50 per cent. less. The manufacturer who would scorn to ride in a vehicle which did not possess absolute strength and finish in all its parts, and who would not countenance any suggestion that unlicensed vehicles should ply for public hire, does not hesitate to erect in his manufactory the cheapest lift that he can buy, knowing also at the same time that the elements of safety are not provided for in the common class of lift. Government inspection should protect the workpeople when the indifference of the employer fails to do so.

In our description of lifts, we shall divide them into the two before-mentioned classes, viz., direct-acting or ram lifts, and suspended lifts. These two classes are often spoken of according to the kind of balance employed, as a weight-balanced ram lift, or hydraulic-balanced ram lift. There are four leading styles of balancing arrangements in vogue for lifts; the two styles most often used are known as the dead weight and the hydraulic balance, while the two less frequently used are the combined weight and compensating balance and the combined hydraulic and compensating

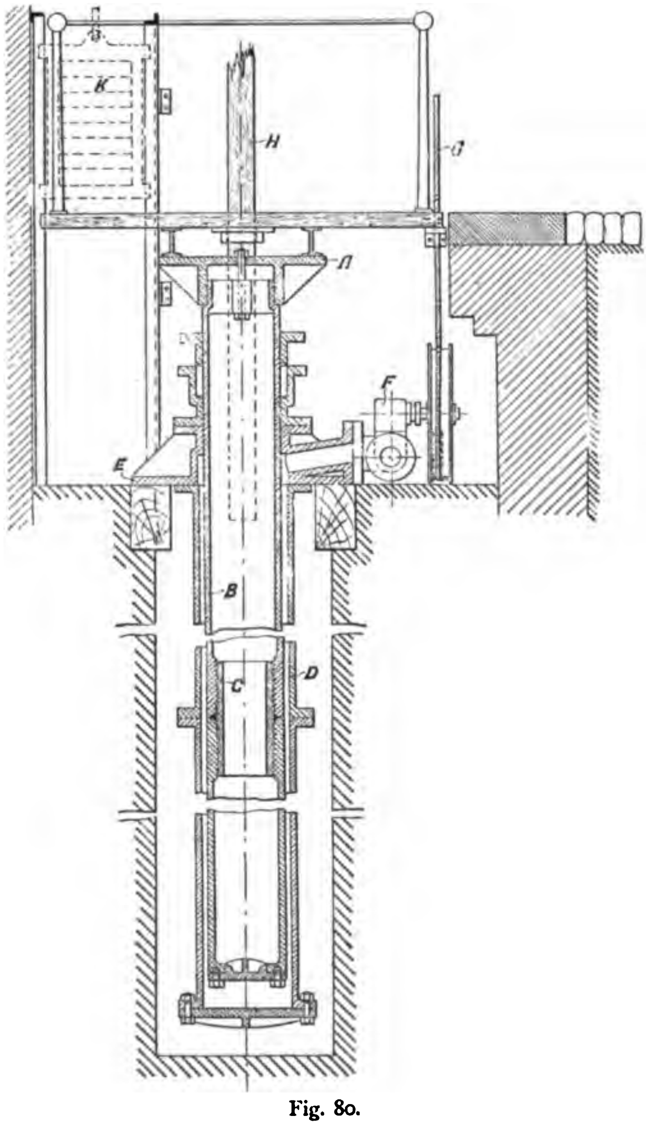


Fig. 80.

balance, the word compensating being used to indicate that the balancing arrangement provides for the varying water displacement of the lift ram while moving in or out of the cylinder.

The conditions that determine the description or style of lift most economical to adopt to meet given requirements are in themselves of such a varying nature as not to admit of classification, depending as they do upon the weight to be lifted, the nature of the weight, the height of lift, the kind of building it is to work in, the nature of the ground the building stands upon, the water pressure at the basement of discharged level, also whether the lift can be worked by an engine and pumps. Generally loads of from 3 tons and upwards are most conveniently dealt with by a ram lift; for lighter loads a suspended lift may be used. It is not usual to put a compensating balance to suspended lifts or ram lifts of short travel, but they are of great economy in a ram lift of long travel, say from 30 feet and upwards, especially when the working pressure in the lift cylinder is small.

Figs. 80 and 81 are a sectional elevation and plan of a dead-weight balanced ram lift for a warehouse consisting of a wooden platform with guard rail upon three of its sides; the platform is bolted to joist or girder iron, and mounted upon a cast-iron platten A. The platten is strongly bolted to the end of a truly turned and polished ram B, made up in lengths of cast-iron piping joined together by screwed nipples C, the pipe ends being tapped to receive them. A blank flange is bolted to the end of the last length of piping to form the end of the ram. The cylinder is made by bolting together pipe lengths D, with a blank flange at the end, the upper end being bolted to the foundation plate E, which is cast with a recess forming an annular space round the ram in excess of that between the ram and cylinder. The foundation plate is provided with a flange to which is bolted the stuffing box, and it also carries the branch to

which can be attached, in most cases direct, the valve F. The rope G from the valve wheel passes round pulleys and up each front corner of the well-hole. Clips are attached to the rope at positions near to the highest and lowest positions of the ram against which a striking bar connected to the lift platform can act, so that when the ram nears its extreme position at the top or bottom of its travel the valve is automatically closed to pressure or exhaust respectively. The

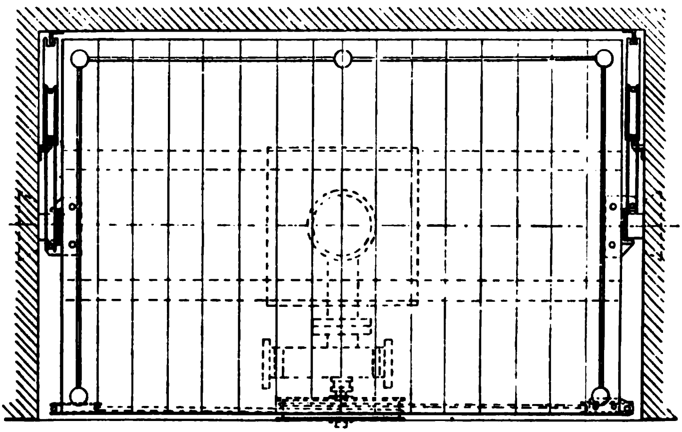


Fig. 81.

slippers or runners which work against the guides are generally cast iron, made an easy fit upon the guides H, which may be made of hardwood, planished bar or T-iron, and are firmly secured to the walls of the well-hole. The adjustable balance weights K are placed in cast-iron frames. These frames run upon T-iron or other guides bolted to the wall of the well-hole, and are connected to the lift by means of wire ropes or chains passing over pulleys on opposite sides at the top of the well-hole.

It is convenient at this point to call the attention of the

reader to a few points in lift design, which materially affect the working arrangements when a load is wheeled on to the platform of the lift; the weight first comes upon the edge of the platform, tending to tilt it, the ram resists this tilting action by a bending stress on cross-sectional planes, and the resistance of the ram to cross breaking ought to be some

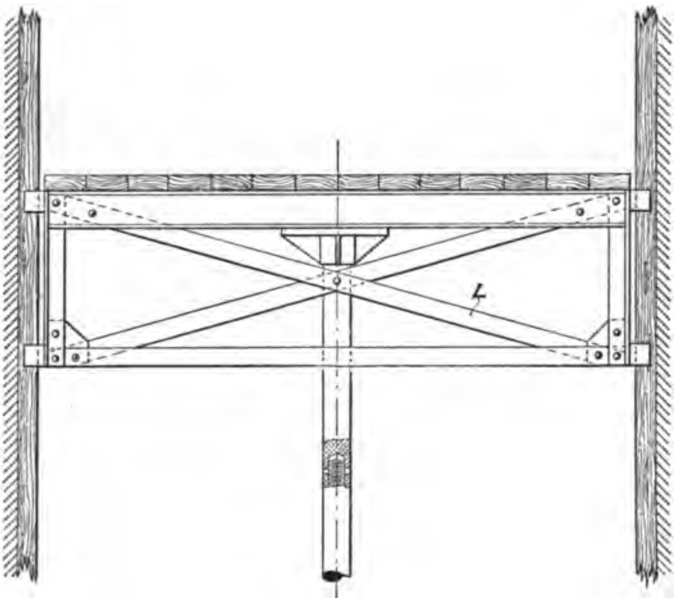


Fig. 82.

six to eight times as much as the stress induced by placing the whole load lifted at the most distant edge of the platform.

Assuming the working pressure to be high, and the ram consequently small, the size of the ram would be insufficient to resist the bending stress induced by the tilting of the platform, and a wrought-iron braced framing L (Fig. 82)

must be provided to carry the platform, having the guides placed close to the top and bottom of the framing. The tilting of the platform is now resisted by the guides, leaving the ram to support the dead load only.

When a cage or cabin is used in place of a platform, this braced iron framing is not needed, the bracing in the cage or cabin being sufficient to prevent bending of the ram.

In making a long ram, by jointing together 10 or 12 foot lengths of piping, the connecting nipples should be so screwed as to leave some 3 inches in the middle of their length plain, and the inside thread at the end of the pipe lengths should be turned off for a distance of $1\frac{3}{4}$ inches from the end, and made a good fit on the unscrewed part of the nipple. After screwing the pipe lengths together, the ends of each length should be drilled, the hole rhymered, and a steel pin driven or screwed in to prevent the nipple from unscrewing.

For the purpose of calculation, the diameter of such a ram built up with lengths of pipes, and considered as a long column supporting a load, may be taken very approximately as equal to half the sum of the diameter of the nipple at the bottom of its thread and the out-diameter of the pipe of which the ram is made. In small diameter rams, as shown in Fig. 82, the screwed nipple is turned out of the solid ram, and its diameter may be .66 to .70, the diameter of the ram. If therefore it is required to ascertain the supporting strength of the ram as shown in Fig. 82, the equivalent diameter of a long solid column of equal strength would be $\frac{1.66}{2}$ to $\frac{1.70}{2}$ or .83 to .85 times the diameter of the jointed or built up ram.

In many ram lifts the pressure or junction pipe from the valve connects direct to the side of the cylinder, and in order that the full waterway of the pipe may be utilised the clearance between the ram and the cylinder should not be less than quarter the diameter of this junction pipe; thus with

a 2, 3, and 4 inch junction pipe the clearance between the ram and the cylinder requires to be ½, ¾, and 1 inch respectively. A 1-inch clearance makes a very large cylinder, and as ½-inch clearance is sufficient for all rams of medium size, and of any run out, it is most economical to cast an enlargement or recess round the bore-hole at the bottom or underside of the foundation plate, and to connect the pipe from the valve to this recess as in Fig. 80.

The size of valve suitable for a medium-pressure ram lift need never exceed one quarter the diameter of the ram, and when the diameter of the junction pipe between the valve and the cylinder is in this proportion the velocity of the water in the pipe is sixteen times the velocity of the ram. In any direct-acting lift when the ram is down, the water pressure acting on the ram is greater than when the ram is up by a column of water equal in amount to the displacement of the ram, and as the ram rises this column lessens by the amount the ram has risen. We will assume an allowance of 1 foot per second as the speed of the ram in the final part of its up stroke, or when it has nearly completed its run out, the platform being weighted with its full load, and the head of water absorbed in overcoming frictional resistances in the pipes and valve, and in imparting the velocity to the water as 12 feet. This is most conveniently allowed for by reducing the working pressure by 5 lbs. per square inch when calculating the size of the ram, therefore in our examples we shall assume 5 lbs. as equivalent to the head of water absorbed in frictional and other losses.

When the high velocity of the water in the pipe joining the valve to the cylinder is considered, it is not surprising that the too sudden closing of the valve to pressure induces vibratory stress in the water, and consequently in the ram, giving the latter jerks or shocks when stopping. It should be the aim of every lift-maker to so construct his lifts as to reduce to a minimum these jerks, especially in lifts for hospitals and hotels.

The best preventative to jerks produced by closing the valve to pressure is to bolt the valve direct on to the cylinder. On the majority of lifts this cannot be done, therefore the connecting pipe between the valve and cylinder should be as large in diameter and as short in length as possible, hence a 2 or 3 inch valve requires a 3 or 4 inch connecting pipe.

To further reduce shock, the port-holes in the valve should be made with V-shaped openings so as to admit of very gradual opening or closing as described in Chapter VIII., while in large valves for low pressure it is advantageous to insert in the valve body a bye-pass valve to act as a shock valve to reduce the intensity of the shocks or jerks of the ram. Some designers arrange an air vessel on the connecting pipe between the valve and cylinder, which will also reduce the intensity of the shocks of the ram, but nothing in the shape of shock valves, or air vessels, is so effective as making the lift valve to give a very gradual opening or closing of the port-holes, while connecting it to the cylinder by a large diameter pipe of very short length.

It is not usually considered necessary to apply safety gear to ram lifts, as the only time an unbalanced ram lift could fall at a dangerously rapid pace would be in the unlikely event of the bursting of the cylinder, junction pipe or valve. This contingency should be impossible if the usual liberal margin of strength or factor of safety is adopted, and the pipes so protected that they cannot be damaged by falling weights. Drain cocks to the cylinder, pipe and valve, to drain off all the water in frosty weather, or for repairs, should always be provided.

The ram of a direct-acting ram lift, either unbalanced, or with a hydraulic balance, acts as a column in supporting the load, and is in compression, but if we attach to the ram platten or platform, by means of wire rope or chain, balance or counterpoise weights, an altered condition of stress is set up in the ram. For a considerable portion of its length

from the top, the ram, instead of supporting the load as a column, is in effect really hanging or suspended from it. Part of the ram is always in tension, and another portion is always in compression, while the neutral or dividing plane, where the tension ends and the compression begins, is constantly varying in position according to the pressure on the ram. Should the ram from any cause become cracked, and thus break above the neutral plane, or should the means of connection securing the platform to the ram give way, then the platform would be violently dragged up to the top by the balance weights, and serious damage, of course, would result. An accident of this character happened to a lift at Paris, where several passengers were crushed to death.

This accident has had a great deal to do with the movements which have been initiated by some inventive engineers to prevent the possibility of such partings of cage and ram; although it is very much to be doubted whether our English practice of firmly constructing ram lifts could even have given room for such an accident.

The application of high pressure to direct-acting lifts is a matter which produces great economy in their working, seeing that but small and slender rams are capable of carrying a comparatively heavy load. These small rams at first give rise to a suspicion of weakness and danger, but from the examples to be seen on every hand working, particularly in London in connection with the London Hydraulic Power Company, we can easily prove their strength, and thus obtain confident assurance of their fitness for the duties they have to perform. Messrs Easton & Anderson supplied a lift for Queen Anne's Mansion, Westminster, where a 5-inch diameter of ram, having a stroke of 101 feet, is working still with a pressure of water due to a column 142 feet high, or about 62 lbs. per square inch upon the area of the ram. This ram weighs 2,817 lbs., and raises a load less than its own weight; thus the upward pressure upon this ram is the pressure per square inch multiplied by the area of the ram

in inches—that is, 23.7 square inches × 62 lbs. = 1,469 lbs., which is a little more than half the weight of the ram itself.

It seems remarkable upon the first glance that such slender rams can safely support a load when standing so far out of the point of rest, as it were, of the ram, which we appear to imagine as a column; but the fact is the rams are seldom under compression, seeing that they weigh more than the load that they have to lift, together with the surplus weight or preponderance which is necessary to cause them to descend when the cage is empty; consequently the water pressure only serves to relieve the weight of the ram, and not to support it altogether.

In all lifts the ram should be screwed and pinned or otherwise securely fastened to a cast-iron cap to which the joist irons can be firmly bolted, the latter making a support to which the wood forming the platform or cabin can be secured. The wire ropes or chains of the counterpoise weights should be securely attached to the ends of the joist irons, and never in any case to the wood forming the platform, nor to the top or sides of the cage or cabin.

In the following examples—

- R = run out of ram in feet.
- l = length of ram in feet.
- p = nett working pressure in pounds per square inch at top level of cylinder.
- W = load to be raised.
- W_1 = load to be raised including weight of cabin or platform.
- x = diameter of ram in inches.

Then for an unbalanced cast-iron ram lift—

$$x = \sqrt{\frac{2W_1}{1.57p - l}}$$

This is the approximate value of x because, after filling in the valves and solving for x, it must be divided by a suitable

coefficient from Table IV. to allow for the stuffing-box friction, and thus the correct value of x is obtained. It should be noted that in the above formula it is assumed that the weight in pounds per foot run of a finished cast-iron ram does not exceed $\frac{x^2}{2}$. Hollow wrought-iron rams are not so common as cast iron ones, and where their finished weight in pounds per foot run does not exceed $\frac{x^2}{4}$, as they need not, we have for an unbalanced wrought-iron hollow ram lift—

$$x = \sqrt{\frac{4W_1}{3.14p - l}}$$

The value of x thus obtained to be corrected for stuffing-box friction by dividing it by the proper coefficient as in the previous case.

If the ram is of small size, and the weight per foot is represented by x^2 lbs., the formula becomes—

$$x = \sqrt{\frac{W_1}{.7854p - l}}$$

CASE I.—Find the diameter of a cast-iron ram for an unbalanced lift to raise 14 cwt. 50 feet high, water pressure 45 lbs. per square inch, platform to weigh 8 cwt. Here we have $l =$ say 53 feet, $p = 45 - 5 = 40$, $W_1 = (14 + 8)\ 112 = 2464$ lbs., then—

$$x = \sqrt{\frac{2 \times 2464}{62.8 - 53}} = 22.42,$$

for a 22-inch diameter ram the coefficient of efficiency $= .99$, hence $\frac{22.42}{\sqrt{.99}} = 22.55$, the corrected value for x. As this is a little over $22\frac{1}{2}$ inches diameter, we should put in a 23-inch ram. Now this would be an absurdly large ram to employ for only raising 14 cwt. 50 feet high, and our reason for noticing it is to demonstrate the saving of water effected as

this common type of lift gradually approaches in design the more perfect form.

A diminution in the size of the ram can be made as some of the platform and ram weight can be balanced, as shown in Fig. 80; we cannot balance all the weight, as some weight must be left in the ram in order that it may descend in the cylinder and force the water through the valve to exhaust when the lift is being lowered without any load upon the platform. The size of ram for a balanced lift is given by the following formula—

$$x = \sqrt{\frac{W}{.7854(p - .434R)}}$$

After solving for x its value must be corrected for stuffing-box friction as before.

CASE II.—Same as Case I., but the ram and platform to have as much as possible of their weight balanced, as in Fig. 80. Here we have $R = 50$, $p = 45 - 5 = 40$, $W = 14 \times 112 = 1568$—

$$x = \sqrt{\frac{1568}{14.36}} = 10.428.$$

On referring to Table IV. we find the efficiency of a 10-inch ram working through a stuffing box $= .98$, hence the corrected value of $x = \frac{10.428}{\sqrt{.98}}$

As this is the diameter of the ram on the assumption that there is no friction in the balance ropes and pulleys, the diameter of the ram as found by the above rule must be increased to allow for the packing in the gland being screwed unnecessarily tight and for the friction of the balance-weight ropes, or chains, over their pulleys, for which we will add 20 per cent. to the ram area, giving in round numbers an $11\frac{1}{2}$-inch ram. The amount of counterpoise or balance weight required is equal to the weight of the ram and platform, less the weight of the column of water displaced by the ram, and the additional allowance to over-

come the friction of the stuffing box, etc., during the descent, equivalent to 10 per cent of the balance weights.

Ram =	3,498
Platform =	896
	4,394
Less water column	2,235
	2,159
Less 10 per cent.	215
	1,944 lbs.

With the water pressure of 40 lbs. the ram would refuse to rise right to the top, but as the lift began to slow down this pressure would rise, approaching the maximum of 45 lbs. A pressure of 42 lbs. is sufficient to send the ram to the top.

Fig. 80 shows a convenient form of balance, as it admits of easy adjustment of the weights.

In the case just considered, the weight of the water column displaced by the ram had to be left unbalanced in order that the ram should descend, and in raising the lift the whole of this dead or displacement weight has to be lifted by the pressure water. In order to obviate this, the balance weight is sometimes connected to the platform by heavy link chain, so that as the ram rises the chain in passing over its supporting pulley at the top of the well-hole gradually increases the weight of the counterpoise, and at the same time reduces the weight to be lifted by an equal amount, and thus balances the water column.

The proper weight per foot of these heavy chain connections is half the weight of the water column per foot. If P represents the pressure on the ram area, W the useful load to be lifted, and w the weight of the water column displaced by the ram—

$$P = W - \frac{w}{2}$$

This result may at first seem paradoxical, as P is evidently less than W, but it is the same as if the pressure acting on the ram is represented by the head acting on the ram at half stroke, thus—

$$P + \frac{w}{2} = W.$$

The diameter x of the ram is given by the following equation—

$$x = \sqrt{\frac{W}{.7854\,(p + .217\,R)}}$$

The ram area given by the above equation must now be increased by 66 per cent., to allow for stuffing-box friction and the friction of the chains and wheels, and a margin to cause the lift to descend empty.

The balance weights must be the same as the total weight of the ram and platform, less the weight of the compensating chains and 10 per cent. for friction and margin to cause the lift to descend empty.

CASE III.—Same conditions as Case I., but with a compensating balance.

$$x = \sqrt{\frac{1568}{.7854\,(40 + .217 \times 50)}} = \sqrt{39.3} = 6.25''.$$

Add 66 per cent. to area and $x = 8''$.

Balance weights: Ram =	1,700
Platform =	896
	2,596
Less compensating chains $\left(\frac{w}{2}\right)$	531
	2,065
Less 10 per cent.	206
	1,860 lbs.

These weights leave a margin of 235 lbs. to overcome friction when ascending with full load, and 205 lbs. when descending.

When the weight to be raised is heavy and the available working pressure small, the size of the ram, balance weights and chains, and overhead wheels or chain pulleys, becomes very large and clumsy. For large weights it is advisable to use an intensifier, and by loading its ram with weights, to convert it into a hydraulic balance. Such a machine is shown in Fig. 83, in which A is a hollow ram, sliding over the fixed ram B, and working in the cylinder C. To the ram A can be attached the adjustable weights F, and the fixed ram B is tied to the cylinder C by the guide bolts G G. The inside of the ram A communicates through the opening D direct with the lift cylinder, and the displacement of the ram B is of sufficient capacity to contain the displacement water when the lift ram is down. The valve is connected to the cylinder C at E, and sufficient weights are placed at F to prevent the lift ram descending too rapidly. When the lift ram is down the displacement water fills the inside of the hollow ram A, which is then quite home in the cylinder C, and upon opening the lift valve the pressure enters the cylinder C, forcing the ram A out, and consequently the ram of the lift. As the balance ram A runs out of the cylinder C, its end pressure gradually increases in proportion to the increased head of water. By suitably proportioning the diameters of the lift ram and ram A, the variation of the load to be lifted, caused by the varying water column in the ram cylinder, may be balanced at all parts of the stroke.

The correct diameters for the lift ram and the ram A can be ascertained as follows:—

Let W = net load to be lifted in pounds.
p = water pressure per square inch at level xy.
R = run out of ram in feet.
R_1 = run out of ram A in feet.
r = ratio of area of ram B to lift ram area.
x = diameter of lift ram in inches.
y = ratio of area of ram A to area of lift ram.

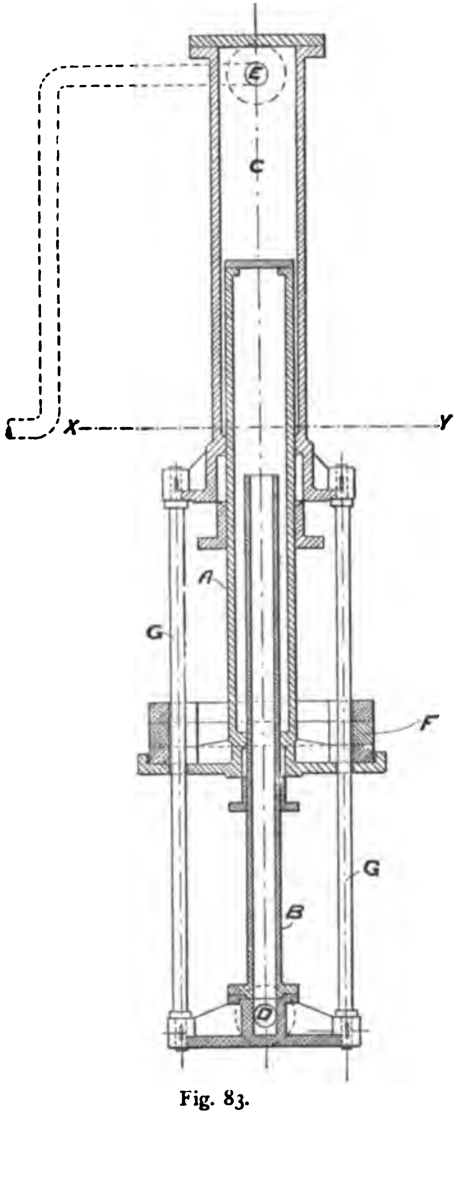


Fig. 83.

In designing, the top level of the lift ram, lowest level of ram A, and exhaust outlet should all be on line X Y. These conditions are assumed in the following equations. The level of the ram A may be varied, but the balance weights F then require readjustment.

$$x = \sqrt{\frac{W}{.7854py}}$$

$$y = \left\{\frac{(R + R_1).43}{\frac{p}{W} + .43R_1}\right\} r.$$

Balance weights = r × (platform + ram − water column). The balance weight thus found must include the weights F, the cylinder A, and the water contained in the annulus between A and B, and lying below the line X Y.

CASE IV.—Same conditions of load and lift as Case I., but to be balanced by the above hydraulic method.

W = 1,568 lbs.
Platform = 896.
R = 50 feet.
$p = 45 - 5 = 40.$
Select $r = 4$.
Then $R_1 = 12.5$ feet.

$$y = \left\{\frac{(50 + 12.5).43}{\frac{40}{1568} + 12.5 \times .43}\right\} 4 = 19.9.$$

$$x = \sqrt{\frac{1568}{.7854 \times 40 \times 19.9}} = \sqrt{10.25} = 3\tfrac{1}{4}''.$$

To find diameter of A, $\sqrt{10.25 \times 19.9} = 14\tfrac{1}{4}''$.

Diameter of B $= 2 \times 3\tfrac{1}{4}'' = 6\tfrac{1}{2}''$.

Allowance must now be made for friction, and diameter of A increased accordingly.

Friction of lift ram : W =	1,568	
Platform =	896	
Ram, $3\frac{1}{4}''$ diam. hollow, $\frac{1}{2}''$ thick =	760	
	2,224	
4 per cent. =	$\frac{2,224}{\quad}$	88.96
Friction of ram B = $2\frac{1}{2}$ °/。 of	$\frac{2224 \times 4}{4}$ =	45
Friction of ram A = 1 °/。 of	$\frac{6400}{4}$ =	16
Total		150
Friction of rams descending empty =		111
Hydraulic friction of descent =		50
		310

Pressure on ram A = 6400, which has to balance 1568 × 4 = 6272, leaving a margin of 128, hence 200 lbs. must be added, requiring an additional area of $\frac{200}{40} = 5$ inches. The ram A must therefore be increased to $14\frac{1}{2}$ inches.

Balance weights : Platform =	896
Ram =	760
	1,656
Less water column	178
	1,478
	4
	5,912

Owing to the increase of the area of the ram A, a discrepancy of about 25 lbs. occurs, which can be rectified by reducing the balance weights.

The word efficiency as commonly applied to lift work has a very vague meaning; its meaning in this chapter is, however, defined as the ratio of the theoretical quantity of water required to raise the load to the actual quantity the lift consumes. The following table shows at a glance the efficiency of the direct-acting ram lifts in the cases that have just been considered. The theoretical quantity of water at 45 lbs. per square inch to raise 14 cwt. 50 feet high is 75.5 gallons.

Case.	Description.	Gals.	Efficiency
—	Ideal lift or theoretical - - - - -	75.5	1
IV.	Compensating hydraulic balance - -	95.7	.79
III.	Compensating and counterpoise balance -	109.0	.69
II.	Counterpoise weight only - - -	225.0	.339

With higher and more suitable pressures the efficiency of ram lifts averages from .75 to .80 per cent., the latter amount being only met with in lifts of good design and build.

Fig. 84 shows Ellington's hydraulic balance, which consists of a balancing cylinder M, connected by distance bolts end to end with the larger working cylinder N. There is a piston to each cylinder, fitted with a leather packing, and connected by a common rod D, working through stuffing boxes in the cylinder covers. The lift cylinder is connected by the pipe H to the annular space E E, which, when the piston G is at top of its stroke, is equal in capacity to the displacement of the lift ram. The annular area L of the lower piston is sufficient when subjected to the working pressure to lift the net load and overcome friction of both the up and down strokes, whilst the full area of the upper piston G is calculated when subjected to the working pressure to balance the weight of the cage and ram less the friction

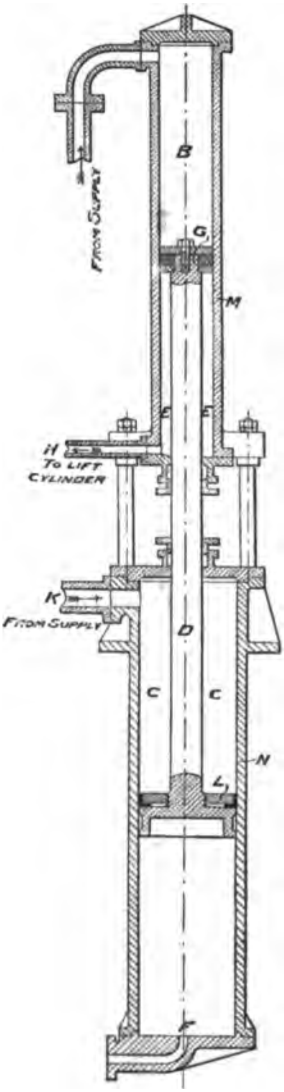

FROM SUPPLY
B
G,
M
E
E
H
TO LIFT
CYLINDER
K
FROM SUPPLY
D
C
C
N
L,
F

Fig 84.

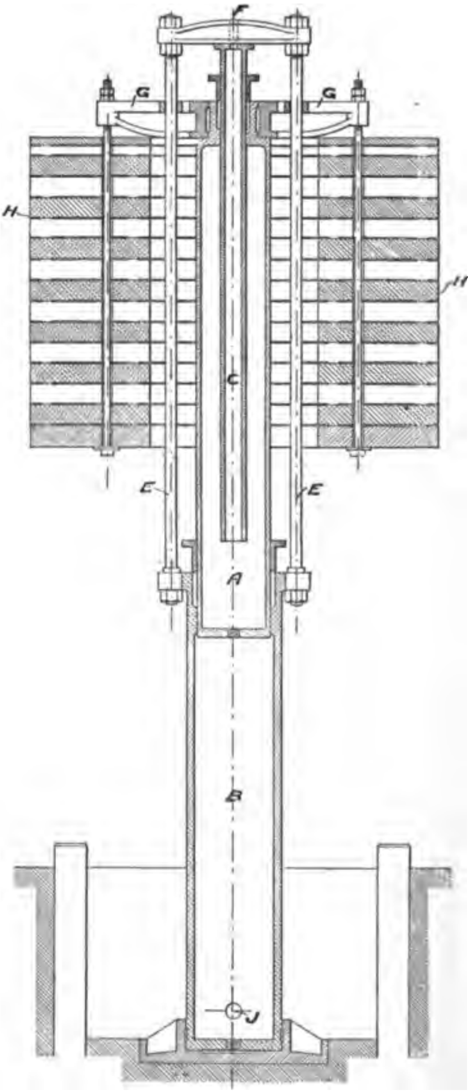

F
G
G
H
H
C
E
E
A
B
J

Fig. 85.

of the down stroke. This piston is subjected to the water pressure at all times.

If the lift ram is assumed to be at the bottom of its stroke, then, on the starting valve being opened, pressure water is admitted to the cylinder C, and the two pistons G and L commence to descend, forcing the water from E E through the pipe H to the lift cylinder; the lift ram is thus caused to ascend, and in doing so requires increasing pressure to compensate for the reduced displacement. This increase of pressure is supplied by the head of water accumulating on the two pistons G and L.

When the exhaust is opened the water from C C only passes away, the water at B being simply forced back into the pressure mains. To make good the leakage the pressure water can be admitted by F under the lower piston when the lift ram is at the bottom of its stroke; thus water will flow from B past the leathers into the annular space E E and supply the deficiency.

If the parts of this apparatus are properly proportioned, the lift ram and the balance pistons are in equilibrium for every part of the stroke. The only serious criticism to be offered to this form of balance is the use of internal packings, it being a *sine qua non* in high-class design to use external packings wherever possible. If in Fig. 83 two inverted rams had been used, in place of the weights F, always open to pressure, an inspection will show that the two systems are practically identical. The lift ram (Fig. 83) would in this case require to be of altered diameter to allow for the weight of water in the two added rams.

When the working pressure is sufficiently high, such as 750 lbs. per square inch as supplied by the London Hydraulic Power Company, it frequently happens that the size of ram required to overcome the load is too small to sustain the load when considered as a column. The hydraulic balance shown in Fig. 85 is much in favour under these circumstances. The water column is unbalanced in this type. A hollow ram A

works through a stuffing box in the cylinder B. The cylinder B is connected by the tension bolts E E to a crosshead F carrying the fixed ram C, working through a stuffing box in the ram A. The ram A is supplied with a crosshead G carrying the weights H H, which are proportioned to balance the dead weight of the cage or platform and the ram, less the water column due to the strokes of the lift ram and the ram A and a margin for causing the down stroke. The cylinder B is connected through the port J with the lift cylinder, and has the same displacement volume. The pressure water enters from the lift valve at F.

When the lift ram is down the balance ram A is up as shown, and on opening the valve the pressure acting on the area of the ram C forces the ram A into the cylinder B, thus causing the lift ram to run out, and when the valve is opened to exhaust the margin of weight in the lift ram to cause the descent raises the balance ram A to the top of the cylinder B. The area of the ram C must be such that, at the pressure available, the total pressure is sufficient to overcome the useful load together with the column of water of a height represented by the stroke of the lift ram added to the stroke of A, and leave a sufficient margin to overcome the friction of the up and down strokes.

When pressure water is not available, either from want of sufficient height or absence of an existing supply, a ram lift can be worked fairly economically by a steam or gas engine, the engine being employed to drive a small pressure pump which forces water from a tank into a small accumulator which has a pipe connection to the lift valve. A suitable pressure for the accumulator is from 1,000 to 1,200 lbs. per square inch, and the capacity of the accumulator should be from one and a half to twice the consumption of water for one complete journey of the lift.

The pumps should be proportioned to deliver when working continuously a larger amount of water than is required by the intermittent working of the lift, and gear

should be fitted such that when the accumulator is fully charged with water the pumps are automatically thrown out of action, thus economising power. A slight fall of the accumulator should bring the pumps again into action.

Where steam power is available the Worthington steam pump can be employed to pump the water direct into the ram cylinder, the valve being controlled by the cord passing through the cage.

The openings to the lift wells in hotels are guarded with light iron gates which the lift attendant alone can open, while in warehouses a wood guard rail is simply hinged to one side of the lift opening. This rail is lifted up when passing in or out of the lift and then dropped upon its supports. Many attempts have been made to secure the opening and closing of the guard rail or iron gate by the up and down movement of the lift cabin or platform, but it is found that mechanical closing begets carelessness on the part of the attendants. By fixing a vertical balanced sliding door in the opening at the bottom of the lift well, and making a hole in the floor to receive the door, the platform or cabin, in its descent, can be made to depress the door level with the floor, and on the ascent of the platform the excess of balance weight will cause the door to rise and guard the well-hole. At the top floor a sliding door can be fixed and partly balanced by means of weights and chains, the top of the cabin or cage being arranged to engage the door in ascending so as to lift it clear of the entrance to the cage, the descent of the cage allowing the door to drop to the floor and guard the well-hole. On the intermediate floors it is most satisfactory to open and close the guard rail or gates by hand.

Passing on to consider the second division of our subject, viz., suspended lifts, Fig. 86 illustrates in elevation the more common arrangement of this form of lift. A is an ordinary cabin or cage, well braced and boarded on the three sides, but open in front. To the bottom of the cabin are secured the two girder irons B lying side by side, with sufficient space

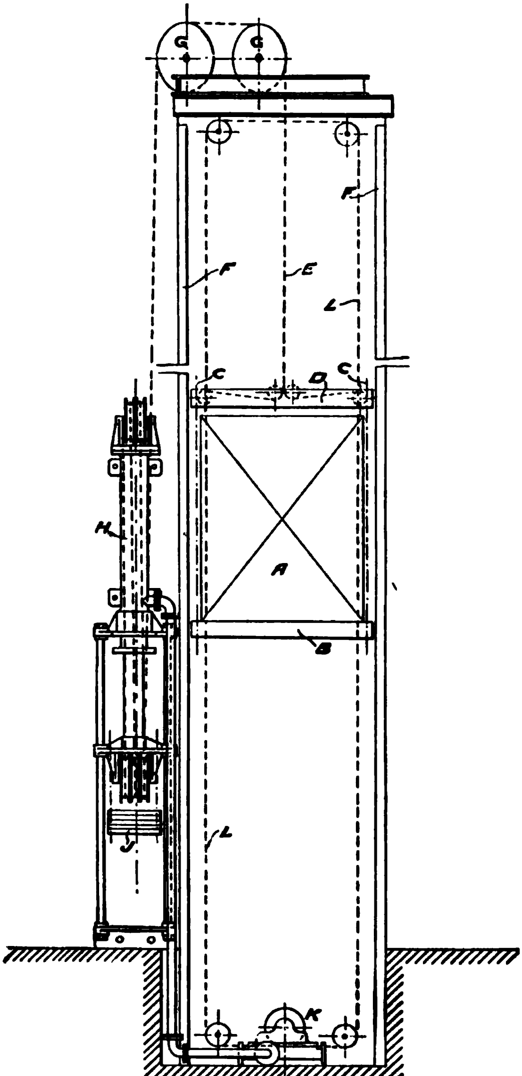


Fig. 86.

between to receive the safety gear. These girders are secured by tension bolts C C to corresponding girder irons D. At the top of the cabin A, and between the girder irons D, are placed the grooved wheels that convey the wire rope to the safety gear fixed below the cage. Two lifting ropes E are used, one passing to the right hand and the other to the left hand of the cage, and thence to the safety gear. Four slipper guides are fixed to the cage, sliding up and down upon the hardwood guides F F, which are securely attached to the brickwork at the sides of the well-hole. The ropes E pass round the overhead pulleys G G to an ordinary hydraulic multiple hoist shown at H. This hoist is made in exactly the same way as those to be described in Chapter XI., and is bolted to the wall with the ram working downwards. To the crosshead are attached the balance weights J, sufficient to almost balance the weight of the cage.

The valve K is placed in the well-hole under the cabin as in the case of ram lifts, and the starting rope passes down through the cage on each side of the well-hole, and is connected to the pulley on the valve. Stops are attached to the starting rope, so that the cabin when nearing the termination of its travel operates against these stops and automatically closes the valve.

The size of the valve need not, as before stated, exceed one quarter the diameter of the hoist ram, and the weight of the ram, crosshead, pulleys, and balance weights should be such as to admit of the cabin descending when empty at the rate of 1 foot per second.

When the cage or cabin is at the bottom level the ram of the hoist is up in the cylinder H, and on pulling the rope L to open the valve to pressure the ram is forced out of the cylinder and the cage ascends until, on nearing the top of its travel, it operates on the upper stop on the rope, thus closing the cylinder to pressure. If the rope is pulled further the cylinder is opened to exhaust, and the excess of weight in the cage above the balance weights causes it to descend,

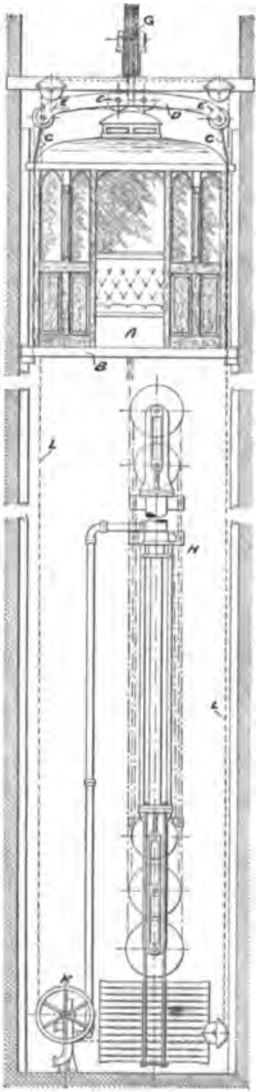


Fig. 87.

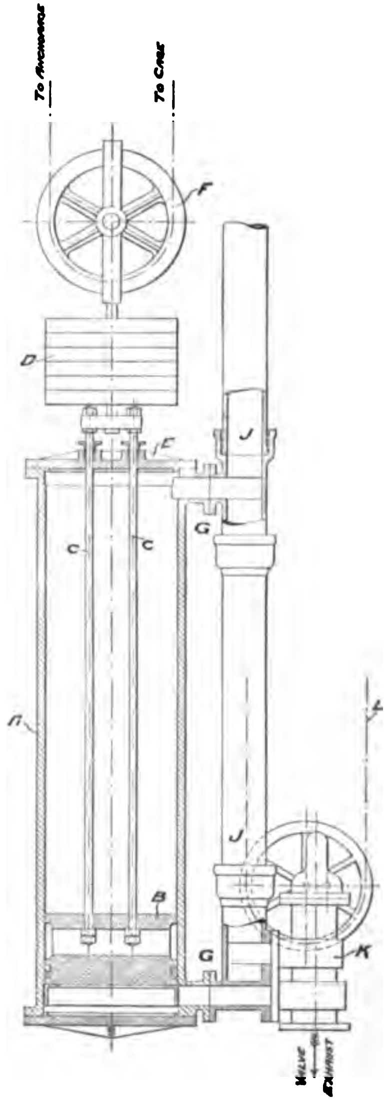


Fig. 88.

pulling the ram back into the cylinder. On nearing the bottom the cage operates on the lower stop on the rope, closing the valve to exhaust. To secure the efficient working of this lift, all the precautions mentioned at the commencement of this chapter must be observed. The correct size of rope and its friction, together with the necessary size of ram for the pressure available, will be considered at the end of this chapter.

Fig. 87 illustrates a high-class passenger lift consisting of a cabin A made of pitch pine, walnut, oak, or mahogany, and having its interior well upholstered and sometimes mirrored. The girder irons B are connected by the bolts C C to the ends of the cross girder D at the top of the cabin. This girder is made in two parts firmly bolted together, and carries the grooved pulleys E, which deflect four supporting wire ropes, two to the right and two to the left of the cage, to the safety gear fixed underneath.

The ropes E pass round the overhead pulley G down to the hydraulic multiple hoist shown at H, which is bolted to the wall at the back of the well-hole with the ram working downwards.

The starting rope L passes down one side of the well-hole through the cabin to the wheel on the valve K, and returns by the other side of the well-hole between the side of the cabin and the wall.

The working of this lift is precisely similar to the one previously described, and the difference in construction of this multiple hoist, viz., placing the rope wheels in line with each other instead of side by side, as shown in Fig. 86, is for the purpose of economising space in the well-hole, and thus allowing a roomy cabin to be used.

The type of hydraulic multiple hoist shown for suspended lifts in Figs. 86 and 87 answers well for water pressures varying from 150 to 1,200 lbs. per square inch; but for less pressures better results are obtained by using the hoist illustrated by Fig. 88, which is largely used.

The arrangement consists of a cylinder A truly bored and fitted with a leather or metallic packed piston B, having two piston rods C C working through hemp packed stuffing boxes in the cylinder cover E, and connected to a crosshead carrying the balance weights D and the pulley F. The cylinder has at each end branches G G. The lower branch connects direct to the valve K, while the upper branch connects to the pressure pipe J, and is not controlled by the valve K. This lift cylinder is generally placed on one side of the well-hole on the basement floor level, and the wire lifting ropes pass from the cage round the overhead pulley at the top of well-hole, and descending pass round the pulley F and upwards to the anchorage at top of well-hole. The action is as follows:—When the piston is at the bottom of the cylinder, as shown, the cabin is at its highest level, and on the valve being moved by pulling the rope upwards the lower branch G is opened to the pressure pipe J, and the pressure water is admitted to the under side of the piston B.

The area of the top side of the piston B is less than the area of the under side by the area of the rods C, hence there is an upward pressure. This pressure, together with the excess weight of the cabin over the balance weights D and piston B causes the cage to descend, lifting the piston B. The water passes from the top of the piston through the valve, and fills the space below the piston. On the cage nearing its lowest position a stop on the valve rope L is operated, causing the valve to be closed.

Upon pulling the rope further, the lower branch G is opened to exhaust, and the water pressure acting upon the top side of the piston forces it down, thus raising the cabin.

Various kinds of safety gear have from time to time been introduced for suspended lifts, many of which are absolutely worthless. Fig. 89 illustrates a well-known type of safety gear suitable for light weight passenger lifts. The hardwood guide A runs from top to bottom of the well-hole, and is en-

gaged by the slipper guides bolted to the sides of the cage. The bracket B is bolted to the under side of the cabin, and carries the tension bolts C C connecting this bracket with the cross girder over the top of cabin. Two bell crank levers D D are pivoted to the angle plate B, their longer arms being

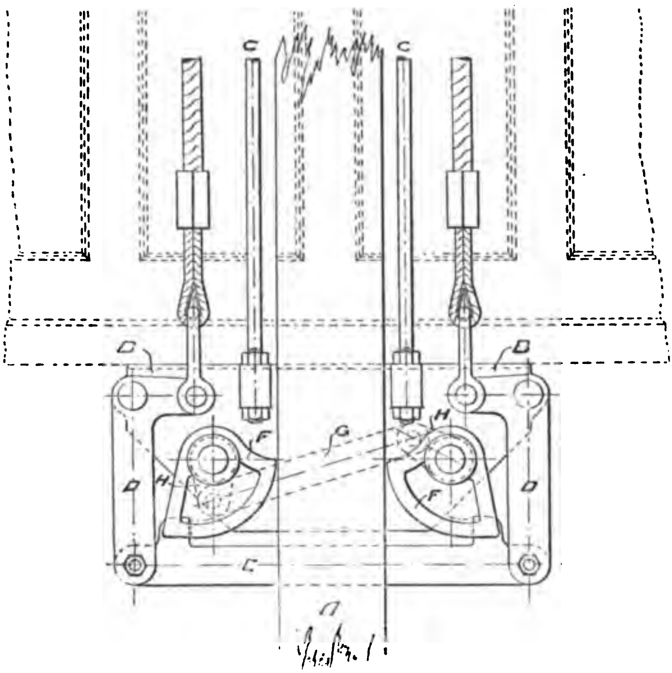


Fig. 89.

joined together by the bar E, which is provided with joggles for engaging the corresponding projections on the eccentric cams F F. These cams are keyed fast upon the ends of two shafts running under the cabin, and supported by bearings formed in the angle bracket B. At the other end of these

shafts two similar cams are keyed, and the shafts being provided with short levers H, that are linked together by the bar G, any movement of one shaft produces a corresponding movement in the other. The cabin should be suspended by four ropes, two of which pass down each side of the cage, as previously described. These two ropes are anchored by means of shackles to the short arm of the bell crank levers D D.

The weight of the cabin is thus divided equally between the four ropes, which are adjusted in length so that the long arms of the bell crank levers D D hang vertically. The cams F F are just clear of the guide A; but upon any one of the ropes stretching or breaking the tension of the adjacent rope pulls the bell crank levers D D out of the vertical, thus pulling over the connecting link E, and causing the cams F F to engage the guide A. The frictional resistance of the cams F F on the guide causes the cams to revolve on their shafts, and firmly grip the guide A, thus supporting the cage. By this arrangement the breaking of any one of the four suspension ropes brings into action the four cams.

Fig. 90 illustrates the Otis safety gear. A is the hardwood guide running from top to bottom of the well-hole. Two rocking levers B are provided, turning on the pins C carried by castings bolted to the wood crossbeams upon which the cabin rests. To the side of the wood beam is bolted the bracket F, carrying the shaft G running under the cabin, and supported at the other end by a similar bracket bolted to the beam. To each end of this shaft are keyed the strikers E, which are actuated by the rocking levers B through the medium of the set screws H H. The cage is suspended by four ropes, two of which pass down each side of the cage, and are fastened to the suspending eyes of the bolts K K. These bolts connect to the lever B at an equal distance on each side of the pin C, and by adjustment of their nuts the lever B is placed horizontally. Should one of the ropes stretch or break while the cabin is travelling up or

down, the lever B, being relieved of the pull of the broken rope upon one arm, is tilted up by the pull of the remaining

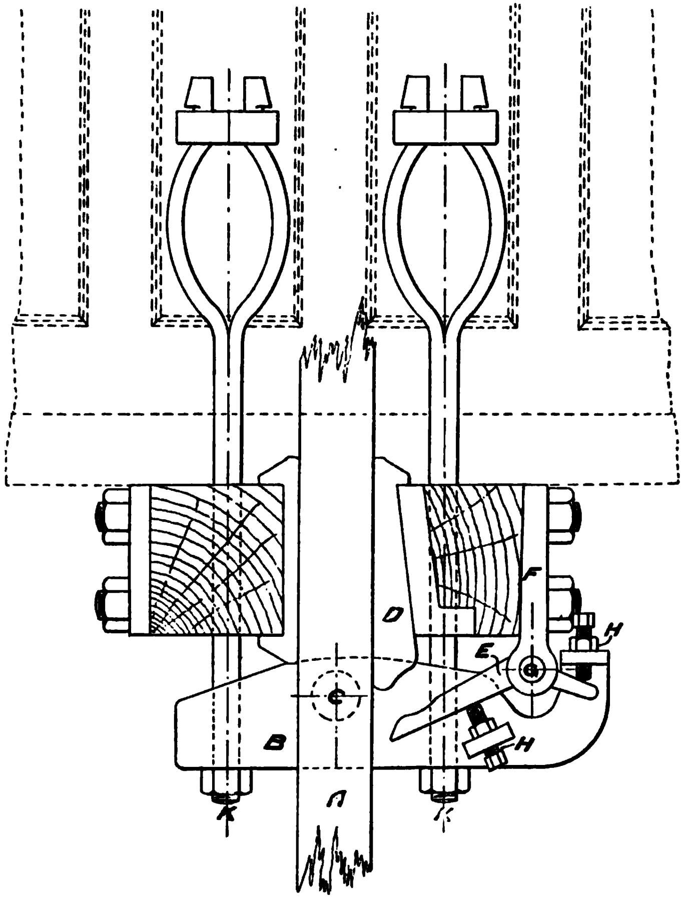


Fig. 90.

rope upon the other arm. This movement of the lever B actuates the striker E, and causes it to push the wedge D up,

thus preventing the further descent of the cage. The friction of the back of the wedge against the casting being much less than that of the face of the wedge against the guide, the weight of the cabin assists in fixing more securely the wedge against the guide.

The two kinds of safety gear described are independent of the elasticity of a spring for their action, and from the fact that they have few and simple parts they are not likely to become clogged with dirt, as often happens with a badly thought out gear.

The number of lifting ropes for suspended cabins or cages varies from two to eight, and as the safety gear requires four it becomes necessary to either increase or reduce the number. This is easily done by introducing a crosshead having three holes for the attachment of ropes. Two ropes are attached, one at each end, and pass off in one direction; while a third rope is attached in the middle, and passing off in the opposite direction, resists the tension due to the other two.

In order to provide against the possibility of a dangerously rapid descent of the cage, due to the valve being opened too wide for the load being raised or lowered, a centrifugal governor, which is actuated by a light endless wire rope or belt suitably attached to the safety gear and passing over idle pulleys, is used. Should the governor revolve too quickly, the rope is retarded by a friction brake, and by the tension thus produced the rope is caused to operate the safety wedges, and check the descent of the cage.

To ensure a long life for the wire ropes of a suspended lift the stress on the wires due to tension, together with the stress due to the wire bending round the smallest pulleys, should not exceed the stress which experience has shown the wire will stand frequently repeated. For steel wire of average quality this stress may be at least 70,000 lbs. per square inch. Again, when life would be jeopardised by an accident, as in a lift or crane, the working stress should not exceed one-eighth the breaking stress of the rope.

The latter consideration will enable the size of the rope to be determined, while by the former the correct size of the wire of which the rope is to be made can be ascertained when the diameter of the smallest wheel over which the rope passes is known. Assuming the breaking weight of good plough steel wire rope to be 150 tons per square inch of metallic section, then the ratio of the diameter of the wires of the rope to the diameter of the smallest wheel round which the rope passes should be about $\frac{1}{400}$.

If the ratio is much larger than this, and the steel of which the wires are made be not of good quality, rapid deterioration of the rope commences, and rupture will take place if the rope is not replaced.

Table VII. gives the breaking weight in tons of good average quality plough steel wire ropes :—

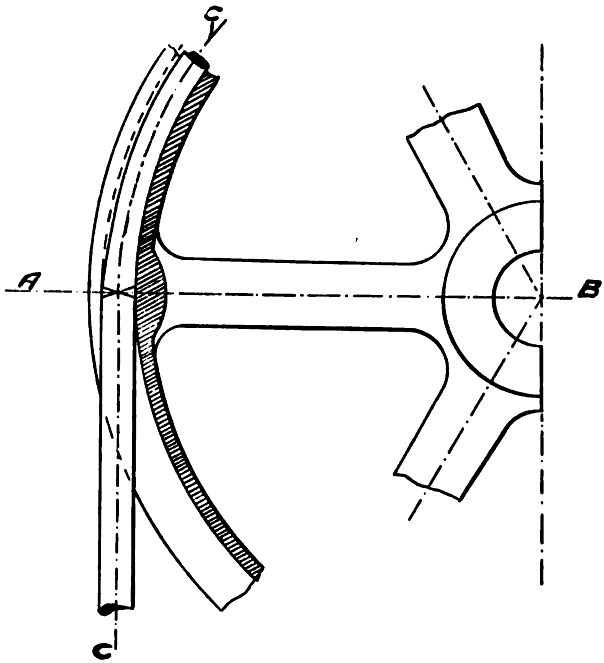
TABLE VII.

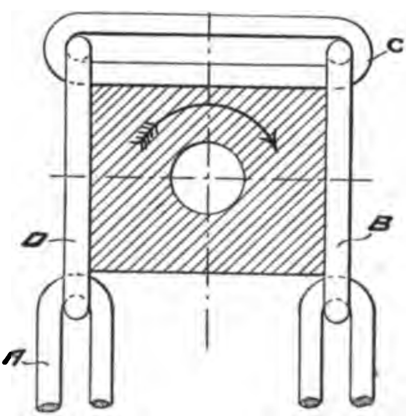
BREAKING WEIGHT OF STEEL WIRE ROPES.

Diam. of rope in inches	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1
Circumference in inches	1	$1\frac{3}{16}$	$1\frac{3}{8}$	$1\frac{5}{8}$	$1\frac{3}{4}$	2	$2\frac{3}{8}$	$2\frac{3}{4}$	$3\frac{1}{4}$
Weight in pounds per fathom - - -	$1\frac{1}{8}$	$1\frac{5}{16}$	$1\frac{9}{16}$	$2\frac{1}{2}$	3	4	$5\frac{1}{2}$	$7\frac{1}{2}$	$10\frac{1}{4}$
Breaking weight of rope in tons - -	4	$5\frac{3}{4}$	$7\frac{1}{2}$	10	$11\frac{3}{4}$	$15\frac{1}{4}$	$21\frac{1}{2}$	$28\frac{3}{4}$	40

When the maximum stress induced in the wires of a rope passing round a pulley does not exceed 70,000 lbs. per square inch, the power expended in bending the rope on to the pulley is largely given off again upon the rope leaving the pulley.

Fig. 91 illustrates part of a grooved rope wheel, and A B is a horizontal line passing through the centre of the wheel, and C C is the centre line of the wire rope passed round the wheel as shown. It is assumed that this centre line does





Figs. 91 and 92.

not alter in length when the rope is bent round the wheel. This erroneous assumption does not perceptibly affect the results. Thus it is evident that the wires below the centre line C C of the rope must accommodate themselves to a less circumference than the wires in a plane normal to the paper; whereas those outside of the centre line accommodate themselves to a larger circumference. The wires accomplish this in the former case by bulging or spreading out laterally and creeping, and in the latter by straightening and drawing in to the centre of the rope. Thus the rope circular below A B before it touches the wheel becomes slightly oval above A B, where it lies in the groove, as shown by the full line D. The distance between D and the dotted line E indicates the extent to which the rope is distorted out of the true circle.

Thus the work lost in bending a rope round a circle is the frictional resistance of the wires sliding upon each other in the act of accommodating themselves to the varying circumferences in which they are forced to lie.

Let D = diameter at bottom of the groove of the rope wheel in inches.

„ d = diameter of the wire rope in inches, and if the coefficient of friction = .2, the efficiency of a rope passed half round a wheel is

$$1 - \frac{.17d}{D+d}$$

The efficiency for various sized ropes passing half round pulleys of different diameters calculated by this formula are given in Table VIII.

Fig. 92 illustrates a square chain wheel with a chain A B suspended from it. In turning the wheel in the direction of the arrow a quarter of a revolution the links A and B each turn a quarter round on their supporting links C and D. Thus when the wheel makes a complete revolution the frictional loss of the chain is the same as that of a link turning

twice round an iron rod of circular section equal in diameter to the bar iron of which the chain is made. Now this holds true whatever may be the size of wheel, pitch of chain, or diameter of chain iron, so that we get for the efficiency of a chain lapping half round a wheel the formula—

$$1 - \frac{.4d}{D}$$

The coefficients of efficiencies for different sizes of chain passing half round pulleys of varying diameter calculated by this formula are given in Table VIII.

TABLE VIII.

COEFFICIENTS OF EFFICIENCY OF STEEL WIRE ROPE AND SHORT LINK CHAIN (FRICTION OF PIN NOT INCLUDED).

Centre to Centre of Rope or Chain across Diameter of Pulley.	Diameter of Rope or Chain Iron in Inches.															
	1/8		1/4		3/8		1/2		5/8		3/4		7/8		1	
	Rope.	Chain.	Rope.	Chain.	Rope.	Chain.	Rope.	Chain.	Rope.	Chain.	Rope.	Chain.	Rope.	Chain.	Rope.	Chain.
2 inches - -	.99	.97	.98	.95	..	..	..	..	..	..	..	..	..	..	..	..
3 " - -	.99	.98	.98	.96	.98	.95	..	..	..	..	..	..	..	..	..	..
4 " - -	.99	.98	.99	.97	.98	.96	.98	.95	..	..	..	..	..	..	..	..
5 " - -	.99	.99	..	.98	.98	.97	.98	.96	.98	.95	..	..	..	..	..	..
6 " - -	..	..	..	.98	.99	.97	.98	.96	.98	.95	.98	.95	..	..	..	..
8 " - -	..	..	..	.98	..	.98	.99	.97	.98	.96	.98	.96	.98	.95	..	..
10 " - -	..	..	..	.99	..	.98	..	.98	.99	.97	.98	.97	.98	.96	.98	.96
12 " - -	..	..	..	..	..	.98	..	.98	..	.97	.99	.97	.98	.97	.98	.96
14 " - -	..	..	..	..	..	.98	..	.98	..	.98	..	.97	.99	.97	.98	.97
16 " - -	..	..	..	..	..	.99	..	.98	..	.98	..	98	..	.97	.99	.97
18 " - -	..	..	..	..	..	..	..	.98	..	.98	..	.98	..	.98	..	.97
20 " - -	..	..	..	..	..	..	..	.99	..	.98	..	.98	..	.98	..	.98
22 " - -	..	..	..	..	..	..	..	..	..	.98	..	.98	..	.98	..	.98
24 " - -	..	..	..	..	..	..	..	..	..	.98	..	.98	..	.98	..	.98
26 " - -	..	..	..	..	..	..	..	..	..	.99	..	.98	..	.98	..	.98
30 " - -	..	..	..	..	..	..	..	..	..	..	..	.99	..	.98	..	.98
34 " - -	..	..	..	..	..	..	..	..	..	..	..	..	..	.98	..	.98
38 " - -	..	..	..	..	..	..	..	..	..	..	..	..	..	.99	..	.98
42 " - -	..	..	..	..	..	..	..	..	..	..	..	..	..	..	..	.99

The formula for the efficiency of a pulley on its axle or pin is the same as for the efficiency of a chain lapping half round a wheel, providing always that the pressure of the wheel upon its axle does not exceed 5 cwt. per square inch (measured on the diameter of the axle), which amount should not be exceeded in lift designing. Table IX. has been calculated by this formula, and for the convenience of readily ascertaining the efficiency of lifts, pulley blocks, etc., the average ratio of the diameter of the pins to diameter of the pulleys is given here :—

Diameter of Wheels in Inches.	Average Pin Ratio.
1 to 16	$\frac{1}{8}$ or .125
16 „ 24	$\frac{1}{10}$ „ .10
24 „ 36	$\frac{1}{12}$ „ .083
Above 36	$\frac{1}{14}$ „ .071

TABLE IX.

COEFFICIENTS OF EFFICIENCY OF PULLEY WHEELS TURNING ON PINS.

Ratio Diam. of Pin to Diam. of Wheel.	.06	.07	.08	.09	.1	.11	.12	.13	.14	.15	.16	.17	.18	.19	.2
Coefficient	.976	.972	.968	.964	.96	.956	.952	.948	.944	.94	.936	.932	.928	.924	.92

To illustrate the practical application of the Tables, let Fig. 93 represent the pulleys in the ram and cylinder crossheads of a hydraulic jigger or hoist, the pulleys being spread out to show clearly the varying stresses in the chain. The top circles indicate the chain sheaves or pulleys in the cylinder crosshead, and the chain is anchored to the cylinder on the right hand, and pays off the left hand top sheave. The bottom circles indicate the pulleys or sheaves

in the ram crosshead, which move downwards in the direction of the arrow.

Let P = total net power forcing out the ram.
„ p = stress on anchorage chain.
„ W = weight lifted.
„ N = number of plies of rope or chain.
„ E = efficiency of pin and wheel with rope or chain round half its circumference.

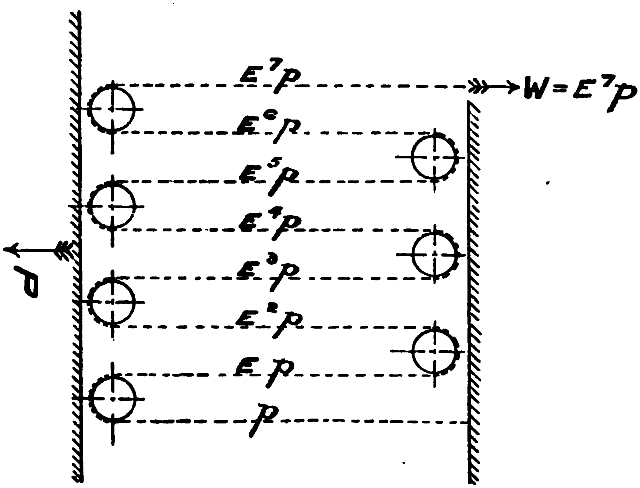


Fig. 93.

When the ram has its full pressure on, but is stationary, $p = \frac{P}{N}$, but the instant movement of the ram occurs some of its power is spent in overcoming the friction of the wheel on its pin and the chain on the wheel. Thus in the figure the stress on the anchorage chain would equal p, the stress on the next chain to it would equal Ep, again on the next to

that the stress would equal E^2p, and so on to the last ply, where the stress would equal $E^{(N-1)}p$. Hence—

$$W = E^{(N-1)}p, \text{ and } p = \frac{W}{E^{(N-1)}}$$

As an example, let the ram of the jigger geared 8 to 1 exert a pressure of $P = 8$ tons, then $p = 1$ ton. The sheaves would be about 14 inches diameter, with about 2 inches diameter pins, so that the efficiency $= .94$. A chain $\frac{7}{16}$ inch diameter would be used, the efficiency of which on a 14-inch sheave $= .98$, therefore the efficiency of the chain wheel and pin $= .94 \times .98 = .93 = E$.

We have $W = .93^7 \times 1$, and this equation is easily solved by means of a table of common logarithms—

$$\begin{array}{r} \log.\ .931 = \bar{1}.96895 \\ 7 \\ \hline \bar{1}.78265 \end{array}$$

The corresponding number of which $= .6$.

$W = .6 \times 1 = .6$ ton. Thus the efficiency of the wheels and chain alone is but .6.

Particular attention is directed to the difference in the stresses of the paying-off end and the anchorage end of the chain. The size of the chain or cable should be determined by dividing the total pressure pushing out the ram by the number of chains or rope plies, and not by merely considering the weight lifted. Many breakdowns in hydraulic hoists have occurred through putting in cable or chain of such a size as only to lift the load safely, and omitting to take into account the extra stress induced in the anchorage end of the chain or cable. In an average hydraulic hoist or jigger geared or multiplied up by pulleys 10 to 1 the stress on the anchorage end of the chain is just twice that on the paying-off end.

Many years ago Sir W. (now Lord) Armstrong published the efficiencies of his multiple hoists, which are very con-

venient for determining approximately the size of cylinder required when the load and working pressure are known. It is advisable to calculate independently the required size of cylinder in each case, only using Armstrong's results given below to aid for first approximation :—

Direct acting	- - - -	93 per cent.
Geared 2 to 1	- - - -	80 „
„ 4 „ 1	- - - -	76 „
„ 6 „ 1	- - - -	72 „
„ 8 „ 1	- - - -	67 „
„ 10 „ 1	- - - -	63 „
„ 12 „ 1	- - - -	59 „
„ 14 „ 1	- - - -	54 „
„ 16 „ 1	- - - -	50 „

CASE V. (see Fig. 86).—Required the size of cylinder for a hoist to raise 14 cwt. 50 feet, working pressure 45 lbs. per square inch. The hoist may be geared 6 to 1, and the travel of the ram is then 8 feet − 4 inches. The height of the top of the cylinder H, above the valve K, is nearly 20 feet, corresponding to a pressure of 8.5 lbs., and the working pressure is 45 lbs. − 5 lbs. (for speed and valve friction) p − 8.5 lbs. = 31.5 lbs.

The hoist has to lift the load of 14 cwt. plus the weight left in the cage to bring it down empty, say 2 cwt., and the area of the ram

$$= \frac{16 \times 112 \times 6}{31.5} \times \frac{100}{72} = 479 \text{ square inches,}$$

the corresponding diameter of which is 24.75 inches.

The size of the rope wheels can now be fixed at 32 inches diameter. Suppose the cabin to weigh 14 cwt., and to be supported by four $\frac{3}{8}$-inch ropes. The first step in the calculation is to determine the efficiency of the ropes working over five 32-inch wheels upon $2\frac{1}{2}$-inch pins. Consulting Table VIII., we find the efficiency of a $\frac{3}{8}$-inch rope on a

30-inch wheel .99, which is also the efficiency of four ropes on the wheel. The efficiency of the sheave on the pin $= .96$. Therefore efficiency of the sheaves, rope, and pins of the hoist only $= (.96 \times .99)^5 = .95^5 = .76$.

The next step is to ascertain how much of the cage weight must be left unbalanced to enable it to overcome the friction to descend empty. The overhead pulleys may be 32 inches diameter, and as the ropes only lap one quarter of the circumference on each, the efficiency of two pulleys is equal to the efficiency of one $= .99$, and the efficiency of sheaves on pins .96.

Therefore, the weight to overcome this $=$

$\{1-(.99 \times .96)\} \times$ weight of cage $= .05 \times 14 \times 112 = 78$ lbs.

Again, the cage has to overcome the friction of the pulley sheaves of the hoist, having an efficiency of .77. Then the weight to overcome this is $(1 - .77) \times$ weight of cage $= .23 \times 14 \times 112 = 360$ „

Finally, the cage has to pull the ram into the cylinder against the friction of its stuffing box, and, from Table IV., this 24 inches diameter ram requires 90 lbs. to move it, so that $\frac{90}{6} \times \frac{100}{77} = 20$ „

Total 458 lbs.

It will be remembered the weight assumed in the trial ram was 2 cwt., whereas we require 458 lbs. As this is the theoretical amount, we must increase it about 10 per cent., giving say 5 cwt. Then we get for the load to be lifted

14×112 - - - - - - - $= 1{,}568$ lbs.

Weight left in cage for descent, 112×5 - $=$ 560 „

Extra pull required to overcome friction of top or overhead pulleys $=$

(load + weight of cage) $\times .05$ - - $=$ 156 „

Total weight 2,284 lbs.

Area of ram—

$$\frac{2284 \times 6 \times 100}{31.5 \times 77} = 565 \text{ square inches,}$$

and correcting this area for the friction of the ram in its stuffing box, $\frac{565}{.99} = 570$ square inches, corresponding to a diameter of 27 inches.

This would be a very uneconomical arrangement of hoist to adopt under the circumstances, but the case is cited to show how necessary it is to independently calculate the size of cylinder required for each case separately, and not trust to any table of efficiencies, as necessarily such tables can only give average results, thus causing the diameter of the cylinder in some cases to be much larger, and in other cases —as, for example, the above—much smaller than required.

CHAPTER X.

WORKSHOP AND FOUNDRY CRANES.

HAVING fully discussed the various valves and lifts worked by hydraulic power, we now proceed to examine the hydraulic machinery used for lifting and conveying heavy weights.

As a fitting commencement of our discussion, we take the hydraulic jack, as being one of the earliest adaptations of the principle of the hydraulic press to practical use. Fig. 94 is a section of the most common type of hydraulic jack. Water is inserted in the cistern or chamber A through the charging-hole B; the screw is now replaced in the hole B, and the jack is ready for use. In working the jack, the head C or toe D is placed under the weight to be raised, and the hand lever E is oscillated, causing reciprocation of the pump plunger F. The water in the chamber A passes into the pump barrel through the suction valve G, and is forced out through the valve H into the hydraulic cylinder I, thus causing the ram K to move outwards in relation to the cylinder carrying the head C and toe D. The ram K is prevented from moving too far out by the small hole L, which allows the water to leak from the cylinder I, so giving a signal that the ram has completed its stroke. To lower the ram the thumbscrew M is loosened, letting the water pass back from the cylinder I to the chamber A.

A few precautions should be observed in working the jack. If the water has been removed for the purpose of examination or repairs, after refilling, the pumps should be given a few strokes with the screw M loose, to force water into the cylinder I, and so drive the air out. When the ram is in use

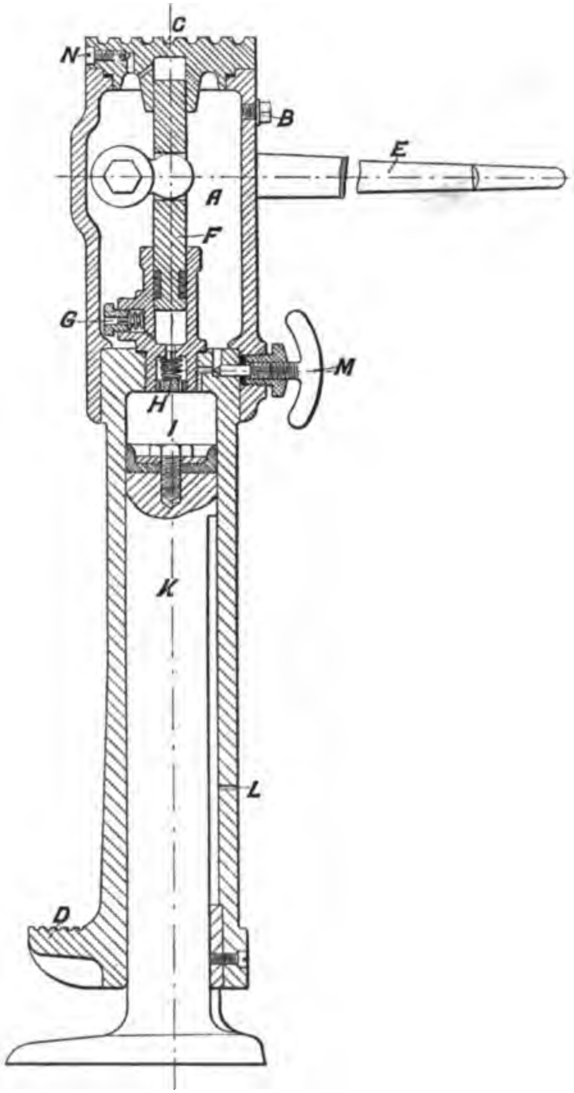


Fig. 94.

the air screw N must be slacked; at all other times it should be screwed home.

Experiments made at various times to ascertain the efficiency of the hydraulic jack give results that agree generally with what might have been anticipated by a theoretical investigation. The accompanying diagram (Fig. 95) shows the general results arrived at by experiment. The ordinates represent the pressures applied to the handle, and the

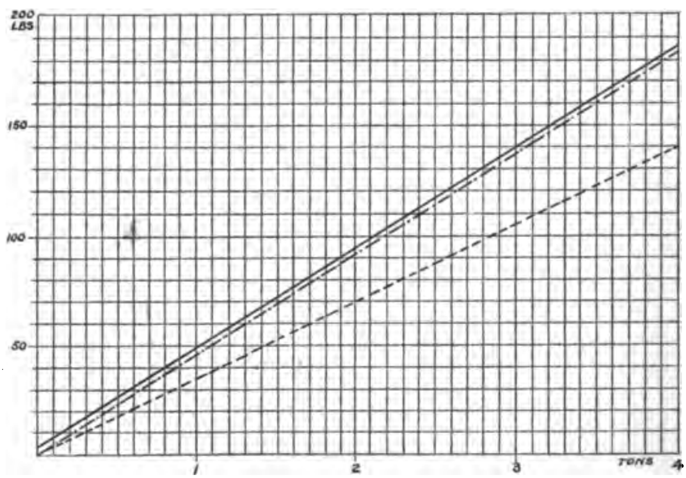


Fig. 95.

abscissæ the loads to be lifted. The full line or "curve" gives the actual pressures required on the handle of a hydraulic jack, having a mechanical advantage of 64 to 1 when lifting various loads. The dotted line gives the pressures which would be required if there were no friction in the jack. An examination of the diagram shows that a pressure of about 3 lbs. is required on the handle when there is no load on the jack, showing that this amount

of pressure is required to lift the jack and overcome the friction.

As the pressure to overcome these resistances will be a constant quantity no matter what useful load is being lifted, we can draw the chain line in the diagram parallel to the full line, and indicating the amount of energy lost on this account. It will now be noticed that there is a further loss to be accounted for, which commences at nothing for no load and increases regularly with the increase of load. This extra loss is entirely due to friction in the various parts of the machine due to the increased pressure on the handle, and consequently increased water pressure. The curve in the figure gives an efficiency of 75 per cent. at the full load, which may be taken as a fair average case in designing, though large jacks in very good condition will show an efficiency approaching 80 per cent. When lifting a quarter of the full load the efficiency falls to about 70 per cent., and for smaller loads the jack rapidly becomes an inefficient machine. It must be remembered that the loss of efficiency we have investigated above is not the total loss, as we have neglected the friction on the up stroke of the handle, also the leakage of the pump plunger and valves.

We are now in a position to fix the diameter of the ram, length of lever, and diameter of pump barrel, so that the only remaining operation is to ascertain the mass of metal required in the various parts to give sufficient strength.

The jack can be damaged by three principal strains, viz.: (1.) The load to be raised by the head may crush the walls of the cistern attached to the jack cylinder. (2.) The load to be raised may shear off the lifting foot at the base of the jack cylinder. (3.) The load may be such that the pressure within the cylinder necessary to raise it may burst the walls of the cylinder. Now the crushing strength of the metal usually employed—viz., malleable iron or cast steel—is so high that the limits of casting actually ensure that the walls will be strong enough to carry the load. We employ, say,

cast steel, which will have an ultimate crushing strength of 40 tons per square inch, or malleable iron, which will have an ultimate strength of 36 tons per square inch; and wishing to make the cistern as light as possible for convenience in handling, we find we cannot get walls to be depended upon in castings which are less than $\frac{5}{16}$ inch in thickness. Thus, if we take a 4-ton jack, our cistern is $3\frac{1}{2}$ inches diameter, which gives us an area of $3.5 \times 3.1416 \times \frac{5}{16} = 3.4$ inches, to carry the load of 4 tons. Other considerations, of course, come in with respect to the arrangement of the metal; but even then the limit of casting ensures us ample margin for safety in working. Similarly, too, the projecting foot, which may be sheared off, is subject to such a light load in proportion to its ultimate strength, that we require to consider chiefly the rough usage which may be given to this projection, and arrange a substantial foot for this, rather than for the actual load to be legitimately lifted by it.

The bursting strain in the cylinder, however, we estimate with more care, seeing that the strain is one of tension instead of compression, and that our metal employed may not be equally strong in each case of straining. The diameter of the cylinder being 2 inches, we have a strain of $2 \times x$ pounds per square inch acting to burst the cylinder, while the metal resisting this bursting tendency is the thickness of the wall on each side; the value of x being that produced by the pump and lever. The load of 8,960 lbs. has to be raised by the pressure given to the 2-inch ram. This amount we have calculated to be 11,636 lbs., and as the area of 2 inches $= 3.14$ inches, the pressure per square inch becomes $11{,}636 \div 3.14 = 3{,}705$ lbs. per square inch in cylinder; $3{,}705 \times 2 =$ total bursting pressure within cylinder $= 3.3$ tons.

Assuming the metal to be of steel, moderately good, and with an ultimate tensile strength of 38 tons per square inch, with a factor of safety of 5, we may put $5\frac{1}{2}$ tons per square inch upon the metal, so that the combined thickness of the walls of the cylinder should equal $\frac{3.3}{5.5} = .6$ of an inch. This

would make each wall $\frac{3}{8}$ inch thick, a dimension which might give trouble in casting in the event of the core slightly shifting, so that $\frac{1}{2}$ inch is allowed instead.

We will next examine some of the more useful designs of workshop and foundry cranes. Fig. 96 shows a very convenient form of wall crane. The ram A is fixed to the bottom of the crane post, and has a hole passing up its centre for the entry of the water. The cylinder B carries

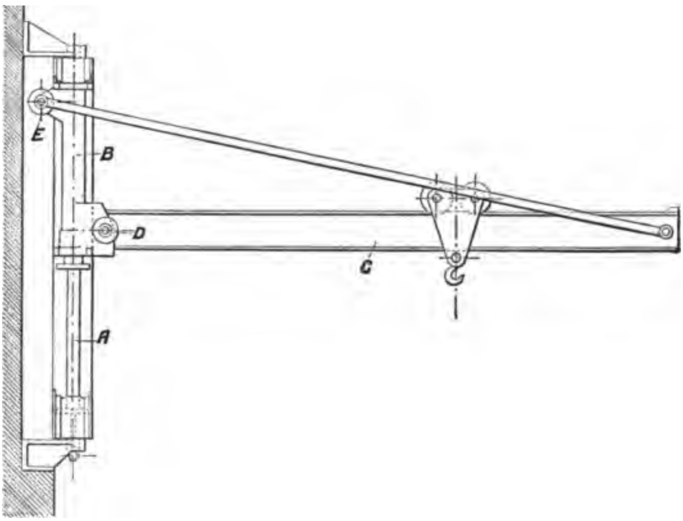


Fig. 96.

the jib C, and moves vertically between the sides of the crane posts so lifting the load, rollers D E being fitted to reduce friction. The crane may be slewed through 180°, the water connection having a swivel joint for this purpose. The valves are placed apart from the crane in a position easily accessible to the workman. This type of crane is generally used to serve machine tools, and is made in sizes to lift from 5 cwts. to 10 tons with a rake up to 25 feet.

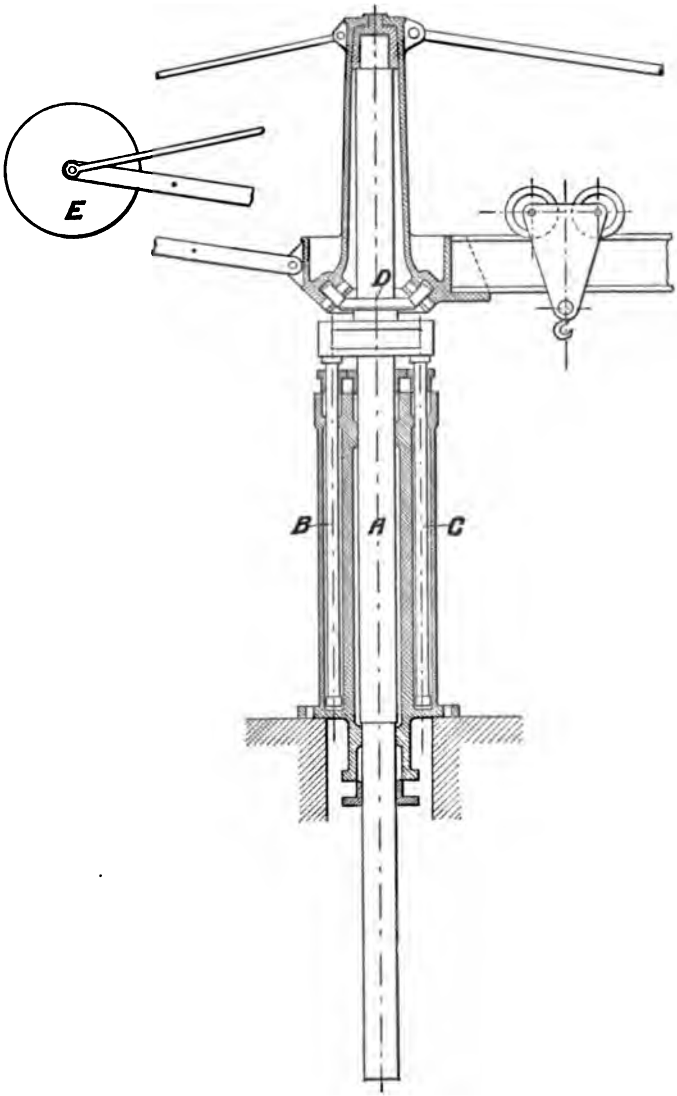


Fig. 97.

Fig. 97 shows a hydraulic foundry crane of a type introduced by Messrs Tannett, Walker, & Co. The large central ram A not only acts the part of a crane post, but has a water pressure always acting upon it by means of the difference of area produced by the reduction of the diameter at its lower part. The total upward pressure acting on this ram is sufficient to nearly balance the total weight of the crane. The two side rams B and C are of a sufficient size to lift the total useful load when brought into use simultaneously. For light loads one only of the rams B and C is used, the other being left open to exhaust. The slewing is operated by hand, the top part of the crane swinging round on the roller path D to reduce friction, while a balance weight E is added which reduces the strains in the crane and also the side friction. This type of crane is very much used in foundries and steel-works.

Another type of crane used in steel-works has a central ram only which is large enough to lift the load and balance the weight of the crane as well; this form is not by any means so economical as the one described above.

For heavy foundry work, the crane as shown in Fig. 98 is employed, having all motions operated by hydraulic pressure. The drawing represents a 10-ton crane having a vertical lift of 8 feet, with a maximum rake of 20 feet. The rams are all fitted with multiplying chains and wheels, so that a short stroke of the ram gives the necessary lift to the load, or motion to the travelling carriage or crane, as the case may be. When water is admitted to the cylinder A, the ram is lifted, the motion being transmitted through the chain B, the travel of which is multiplied in the ratio of 4 to 1 by the pulleys C. This motion is, however, halved by the block D, so that the travel of the weight to be lifted is double that of the ram A^1. The racking motion is performed by two small rams E F, arranged side by side, and having chains attached to the travelling carriage G. These rams are so arranged that when one is fully out the

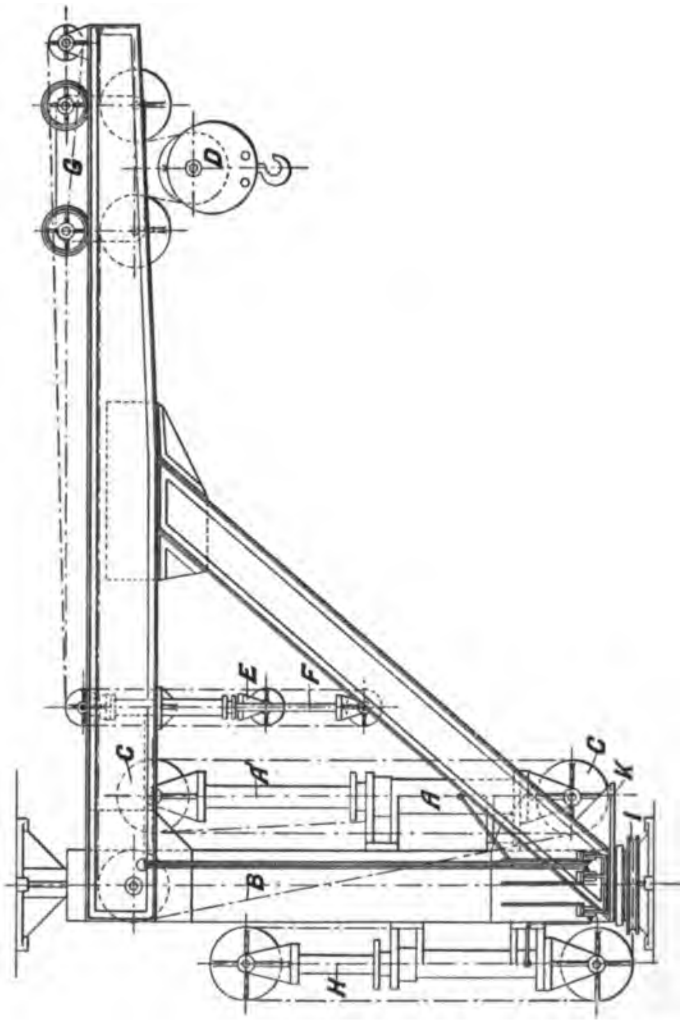


Fig. 98.

other is in. On admitting water to the one that is in, the carriage is travelled or racked along, the other ram being drawn in at the same time. The slewing motion is per-

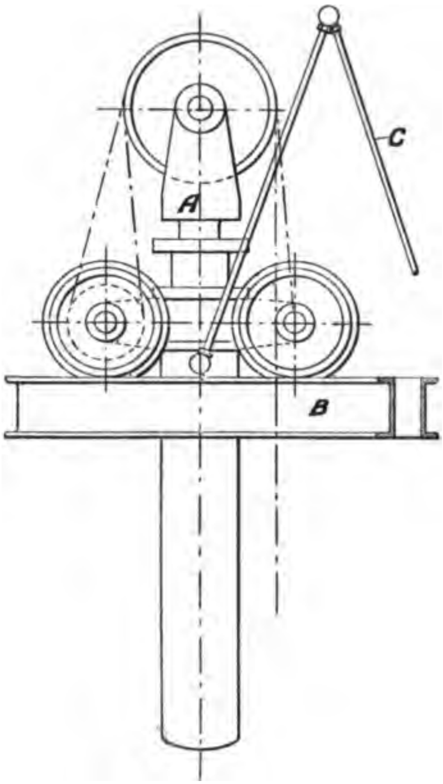


Fig. 99.

formed by two rams H, placed at the back of the crane post, and similarly arranged to the rams E F, but much larger in diameter. These rams travel with the crane and act on a fixed wheel I secured to the floor plate. All the valves are

placed on the side of the crane post, and are operated by an attendant from the foot-plate K. The dead weight of the crane and load is supported by live rollers.

Other types of shop cranes are simply modifications of those described, arranged to suit special requirements. In auxiliary lifting appliances, the handy tool, shown at Fig. 99, is useful for light work, such as lifting weights into and out of lathes or other machines. The ram A is supported on rollers running on channel irons B, which may form the jib of a

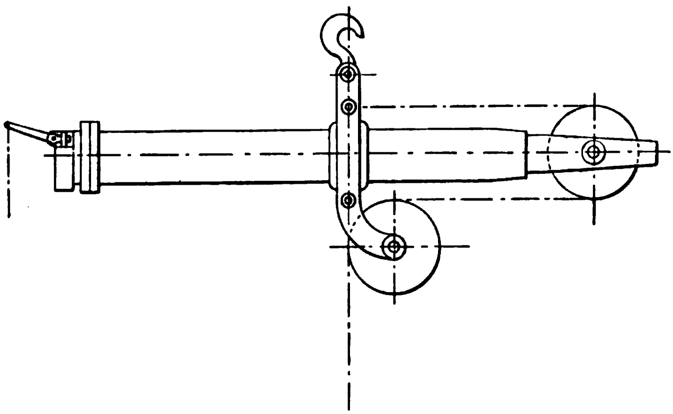


Fig. 100.

crane, or may be fixed over the machine to be served. The water is fed in through the walking pipe C, having swivel connections, the valves being placed near the machine to be served, and handy to the workman. The ram and cylinder are sometimes placed in a horizontal position. This form of lifter is very useful in connection with riveting machines, being used either to support a portable riveter, or the work to be riveted by a fixed riveter.

The form shown in Fig. 100 is intended to be supported from a crane, and carries its own valves, the water being fed

to the valves by a spiral pipe. By the use of one of these the work can be much more quickly and accurately adjusted for riveting than if the large crane is to be operated each time.

Fig. 101 shows a form of direct puller without any chain multiplying gear.

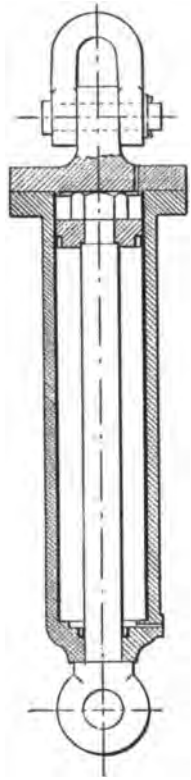


Fig. 101.

The principle of water acting upon a ram or piston is so definite and constant, that it has been applied most ingeniously by Mr Duckham to suspended weighing machines. The application is one that has special advantages for crane or dock work, seeing that the amount of rough usage generally extended to such appliances is quite sufficient to damage any spring, and to damage any lever or elaborate mechanism. The attention given to this class of machinery is such that the gauges or standards are absolutely accurate.

We illustrate the machine in section in Fig. 102. The construction of the machine we will now describe in detail. The cylinder is bored out perfectly true and lapped with emery to a fine dead polish, thus ensuring an absolutely smooth surface; the piston rod B, with its plates and leathers, is then fitted. A is the hanging strap, B the piston rod, D the cylinder, C the space filled by the liquid. The indicator gauge screws into the cylinder, and a filling plug is also inserted in the cylinder, so that it may be filled with the liquid when desired. Oil is generally employed, although in cold climates glycerine is sometimes used. Leakage will not affect the correctness of the indicator upon the gauge unless the

piston actually comes into contact with the bottom of the cylinder, when it will, of course, cease to indicate until

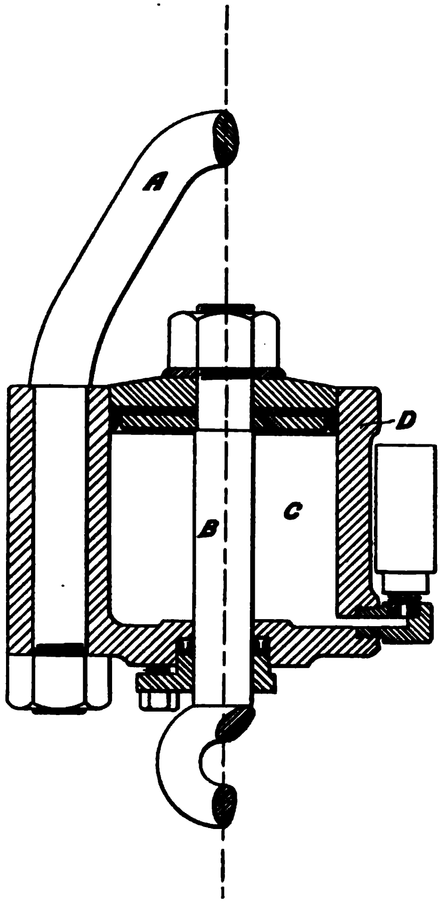


Fig. 102.

filled. Re-filling is usually necessary about once a month when the machine is in constant use.

When a load is suspended from the piston rod of the machine a pressure is communicated to the liquid, which pressure is then transmitted to the indicating gauge for registration on the dial. The gauge is of the ordinary Bourdon type, having an elastic steel tube of a flattened form of transverse section at one end, and bent to present the figure of a circular arc. The effect of the pressure is to flatten the curvature of the tube and to cause the free end to move with an oscillatory motion; the free end of the tube has connected to it a rod which gives motion to a rack gearing into a pinion working the hand which indicates the pressure. These suspended hydraulic weighing machines are now used for dead weights requiring indication up to within 20 lbs., such for example as for weighing boilers, heavy goods, and large packages, where they have been found to be invaluable.

CHAPTER XI.

WAREHOUSE AND DOCK CRANES.

THE importance of this branch of hydraulic machinery will be appreciated when we state that it was to the wharf crane that Lord Armstrong at first applied the hydraulic principle, the pressure being obtained from an elevated tank. The elevated tank, however, soon had to give way to the dead-weight accumulator. The success of the early Armstrong cranes was such, both from satisfactory working and saving in cost, that the system rapidly spread, until to-day it is almost universally employed for wharf purposes.

In some of the original designs internal packing was used in order to provide two powers to the crane; this practice has now been abandoned, and all packing is external wherever possible.

Fig. 103 shows a multiplying hydraulic jigger. This very useful and most frequently employed appliance has the advantage that it can be placed in any convenient position either inside or outside of a building, working vertically or horizontally, and the rope or chain can be led off to raise a cage or for use with a crane jib. A ram A works in the cylinder B, and has a set of pulleys attached to its head, a similar set being secured to the base of the cylinder. The lifting rope or chain is anchored to the cylinder, and passes alternately over the pulleys attached to the ram head and the cylinder base, and finally away to the load, thus multiplying the stroke of the ram. In the illustration the stroke of the ram is 5 feet, which is multiplied 8 times, giving a lift of 40 feet, while the net load lifted after allowing for friction is 1 ton. If the ram is placed horizontally a slightly

larger allowance for friction must be made. Guide rods C are provided to direct the ram A in its outward course, also to act as a stop when the ram has made its full stroke. The valve D is automatically closed at the ends of the stroke by the tappet rod E.

As the loads to be lifted vary greatly, it is often desirable to have more than one power, and so save pressure water. There are two good ways of effecting this, which we will describe. By the first method the cylinder is made larger in bore than the diameter of the ram to lift light loads, and a second ram is used, made in the form of a tube, and carrying a stuffing box through which the smaller ram works. This tubular ram has no base, so that the water has access to both rams. The outer ram works in a stuffing box on the cylinder in the usual way. Now if both rams be left free to move when the water is applied, the lifting effort will be that due to the combined area of the two rams, or in other words, to the area of a circle having a diameter the same as the ram working through the outer stuffing box. This constitutes the higher power. For lifting light loads the tubular ram is secured in its lower or in-position by a pair of claws which are passed over its upper

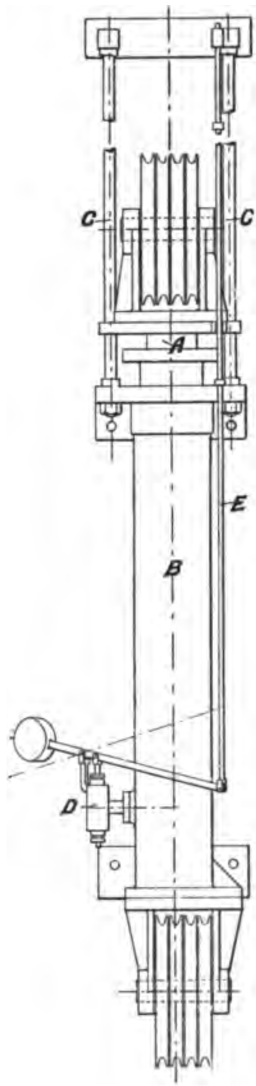


Fig. 103.

edge, so that the water pressure is only free to operate the smaller ram. By the second method three equal-sized rams working in three cylinders placed side by side are all attached to one common head carrying the rope pulleys. By passing pressure water to all three rams, the maximum load is lifted,

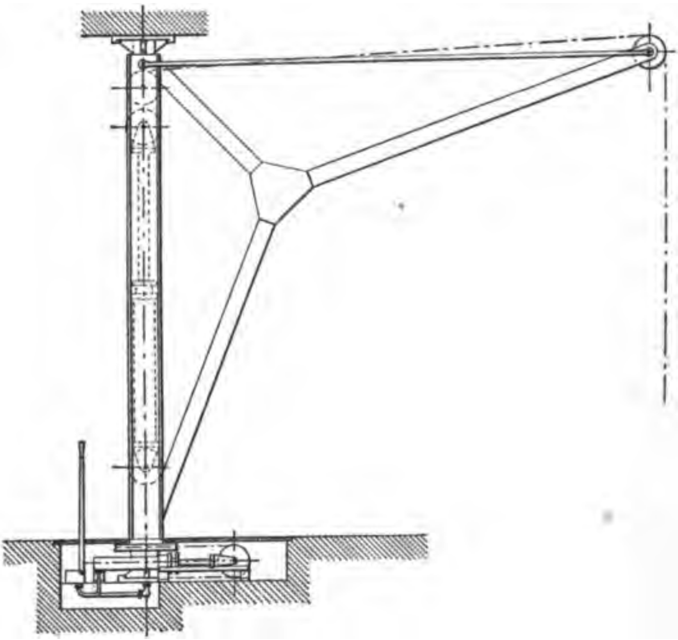


Fig. 104.

whereas if the central ram be opened to exhaust the remaining two will lift two-thirds of the maximum load. For very light loads the central ram only is used, the other two being open to exhaust.

Fig. 104 is an illustration of a crane suitable for use in railway goods sheds, and for general loading and unloading

purposes. The lifting is performed by a multiple jigger of the type already described, while the slewing is operated by two small rams placed under the floor, which alternately pull a chain which is anchored to a pulley upon the pillar. The valve levers are placed at the back of the crane.

Another very common type of warehouse crane is the wall crane used for loading and unloading ships. These cranes are fitted with long jibs having a derricking motion operated by a hydraulic ram, also a slewing motion of 180°, so that one of these cranes can serve a wide frontage of the warehouse.

It is often convenient to employ a travelling wharf crane, such as shown in Fig. 105, which is of the bridge type, having an opening large enough for a railway truck to pass through. The pressure water is supplied from stand pipes or hydrants by walking pipes. The arrangement will be readily understood from an inspection of the drawing. All valves are contained in the cabin.

In another type of travelling wharf crane the base is made short without the bridge, but in all other respects the design is similar to the one illustrated in Fig. 105. These travelling cranes should always be provided with rail clips to grip the rails, and so steady the crane when lifting heavy loads. Screw blocks are also provided on heavy cranes to relieve the axles of the load.

Fig. 106 illustrates a large dock crane capable of lifting 160 tons through a height of 50 feet, with a direct puller of the type already shown in Fig. 101. For lifting lighter loads of 35 tons a 3 fall chain block is used, operated by a hydraulic motor or ram and cylinder. The chain passes between pocketed or pitched chain rollers on the motor, and is then deposited in a well. The slewing is performed by a hydraulic motor, which drives a vertical shaft carrying a pinion wheel gearing into a large circular rack. When it is intended to use the chain hoist the large hydraulic cylinder is drawn into an inclined position by a chain attached to a hydraulic capstan.

The valve of the large cylinder is operated by a man standing on the elevated platform; all the other movements are operated from the cabin. The pressure water is supplied from a plant of machinery separated from the crane.

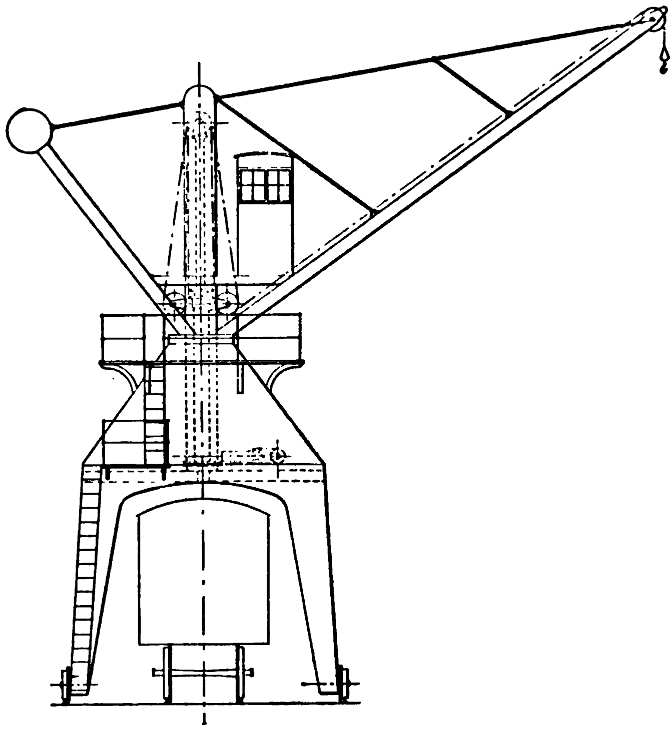


Fig. 105.

We will close our remarks on cranes with a caution respecting shock due to the too sudden closing of valves. If a load is being raised or lowered it has velocity, and therefore kinetic energy; now this energy must be absorbed in

doing work before the load can be brought to rest. The only means we have at our disposal is to close the valve, and so cause a rise of pressure in the hydraulic cylinder. As

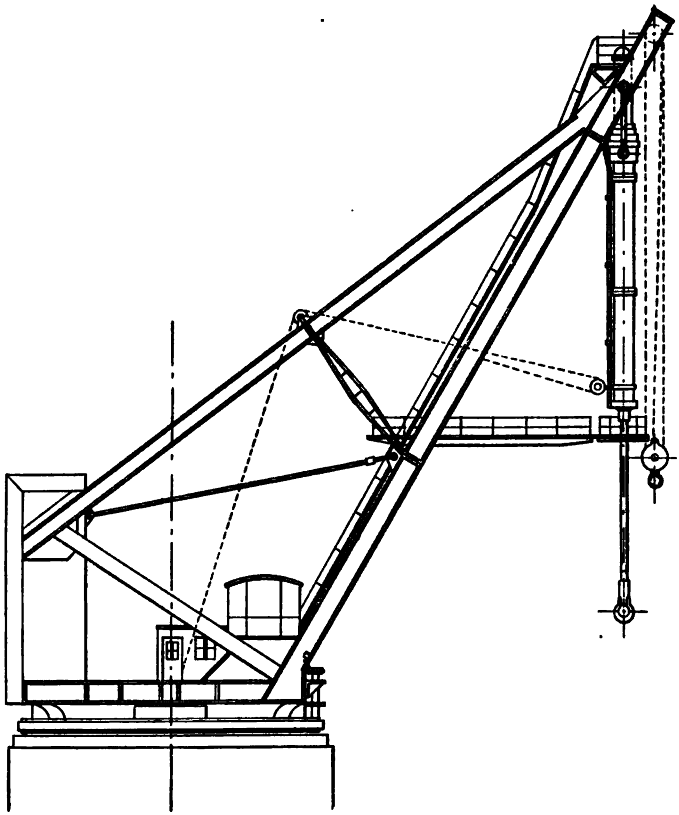


Fig. 106.

water is only very slightly compressible, the load must be almost at rest by the time the valve is closed if there is no relief valve. A knowledge of the laws of moving bodies

tells us that the less time taken to arrest motion the greater is the force or pressure required, so that in reducing the time by closing the valve quickly we greatly increase the water pressure, and a cylinder may thus be broken. By inserting a shock valve either opening to the accumulator pressure or controlled by a spring, we ensure that the pressure in the cylinder can never rise above some fixed amount.

CHAPTER XII.

HYDRAULIC ACCUMULATORS.

HYDRAULIC power is generally employed in an intermittent manner, and when the pressure is produced by mechanical means, the demand upon the pumping machinery is frequently very great, while at other times it may not be required at all for some period. It is thus evident that if the water were to be used direct from the pumps, they would have to be of sufficient capacity to meet the utmost demand, and to be capable of giving the maximum quantity required at all times and periods; so that, in fact, an immense waste of energy would result, owing to the diminished conditions requiring a diminished supply from the pumps.

Thus, supposing for example that a lift and a press are to be supplied with hydraulic pressure by means of a pumping engine, and that the lift requires 100 gallons and the press 60 gallons per minute, a pump must be employed capable of meeting this double demand, and must supply 160 gallons of water per minute. But the lift will not require the water more than once in every five minutes, while the press will require to be supplied once only in every ten minutes, when working at its greatest possible speed. This united demand, then, requires in one minute out of every ten that 160 gallons of water at full pressure shall be supplied with promptitude and certainty; but for nine minutes out of every ten this amount would be considerably in excess of the actual needs, seeing that during every five minutes an absolute cessation of delivery to the lift and the press is thus secured for a period of four minutes. The average amount of water that could be supplied, provided means

were at hand for storing up the quantity ready for the full demand, we can determine very easily. During every ten minutes the lift will have made two strokes, and in so doing will have consumed each time 100 gallons of water. In the same time the press will have required 60 gallons of water.

Thus 260 gallons of water will be required in that time, so that, if the pumps can be allowed to run constantly, they can be set to work with a delivery of 26 gallons per minute theoretically. But to provide for leaks or waste we require, say, 25 per cent. above this amount, and thus supply $32\frac{1}{2}$ gallons per minute for the duty named.

The simplest way of storing up this water is to erect a tank at a height sufficient to give the required pressure by the weight or head of the water column alone. This arrangement is frequently and generally adopted for hydraulic lifts in warehouses, hotels, and lofty buildings. The water used upon such premises for this purpose is usually pumped up over and over again, so that a large amount of water is not required, as the water escaping from the lifts discharges into one common tank, from which the pump draws it again. As soon as the water rises to its determined height within the tank, a ball or other valve closes the delivery pipe, and the pumps stop; and when the water level falls, they again start automatically. With this kind of demand it is absolutely essential that the pumps should start off without any dead centre to be overcome or met, and it is found that no pump will maintain this supply, stopping and starting even after standing for a length of time, so well and so effectually as the Worthington. The advantage of employing a tank for such work as that of supplying a lift is obvious from the fact that water may be pumped up in the daytime, ready for any demand which may be made during the night, while the pumps are themselves not at work.

When pressures such, for instance, as 700 lbs. to the inch are employed, it becomes quite impracticable to adopt a tank or a water tower, seeing that a column to give that pressure

would need to be 1,610 feet high, and pressures as great as 3 tons to the inch of course could not be provided for

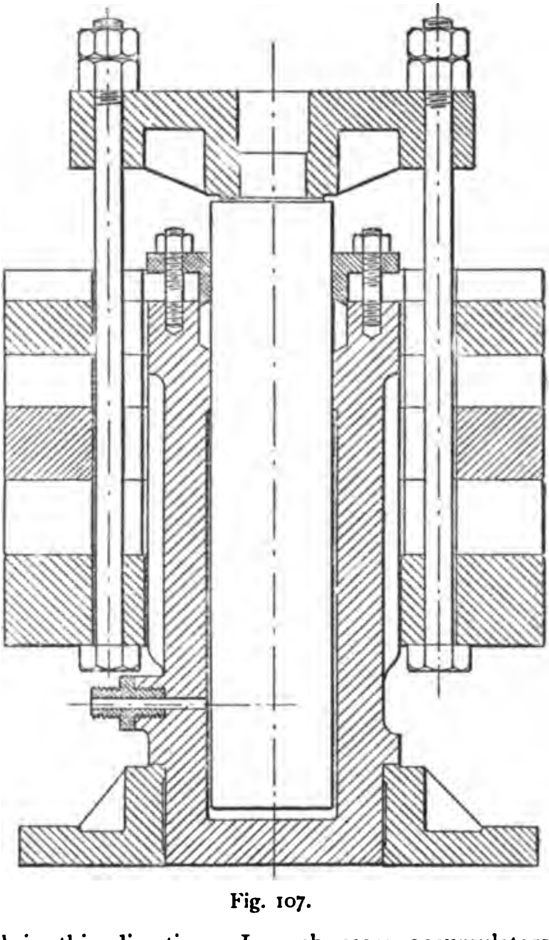


Fig. 107.

at all in this direction. In such cases accumulators are employed, and assume generally the form of a vertical cylinder, fixed at one end, as illustrated in Fig. 107, and

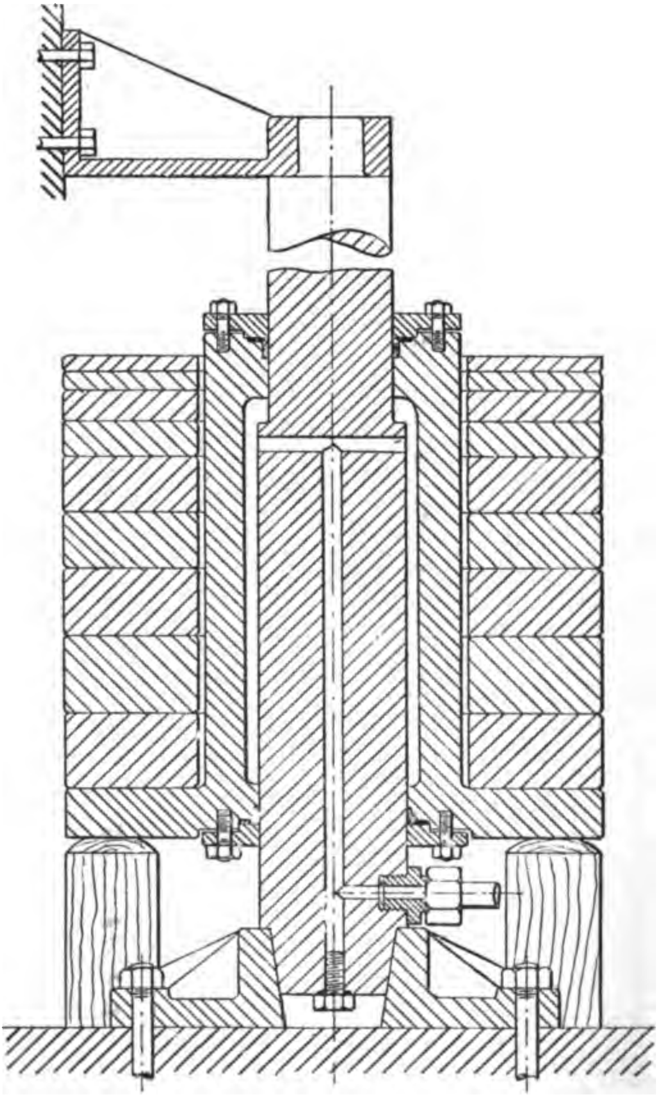


Fig. 108.

free at the other, having a ram or plunger working through a stuffing box and gland, or through a gland and leather cup packing, as indicated in Fig. 108. The hempen packing is the best, owing to its being more easily renewed, but great friction is often induced by such glands being too tightly screwed down. The ram or plunger carries a load, which, in the example illustrated, is made up of cast-iron weights of circular form, which are suspended from the head of the ram cap by means of long bolts passing through them. Instead of cast-iron weights, where space is not so valuable, a tank or vessel, as shown in Fig. 109, is carried by the bolts passing down from the ram cap, either in a truly vertical form, or inclined so as to obtain a more central or distributed support for the load. Within the tank all kinds of material in loose form, such as slag, stones, bricks, etc., are thrown to make up the amount necessary to give the required pressure upon the ram, in order that it may store up the work that the pumps are doing.

The accumulator should be placed as near to the pumps as possible; and if the system of pipes supplied is very extensive, it is often desirable to place another accumulator in some position where it may be of most service in taking up quickly any sudden demand that may be made upon the pipes. The load of the accumulator is made to strike against a stop when quite up, so that as soon as it is lifted to the full height the water cannot escape from the pumps, and they are compelled to stop until the reduction of the pressure by the draught of water permits them to start again. The weights are sometimes arranged to act upon a rod which has a collar attached at any desired point, so that when the weights or the tappet oar strikes the collar the valve is closed, the steam supply shut off from the pump, or the belt driving the pumps is thrown on to the loose pulley. When the weights fall away from the collar by reason of the draught of water from the accumulator, the rod controlling the valve or the belt also falls by its own weight,

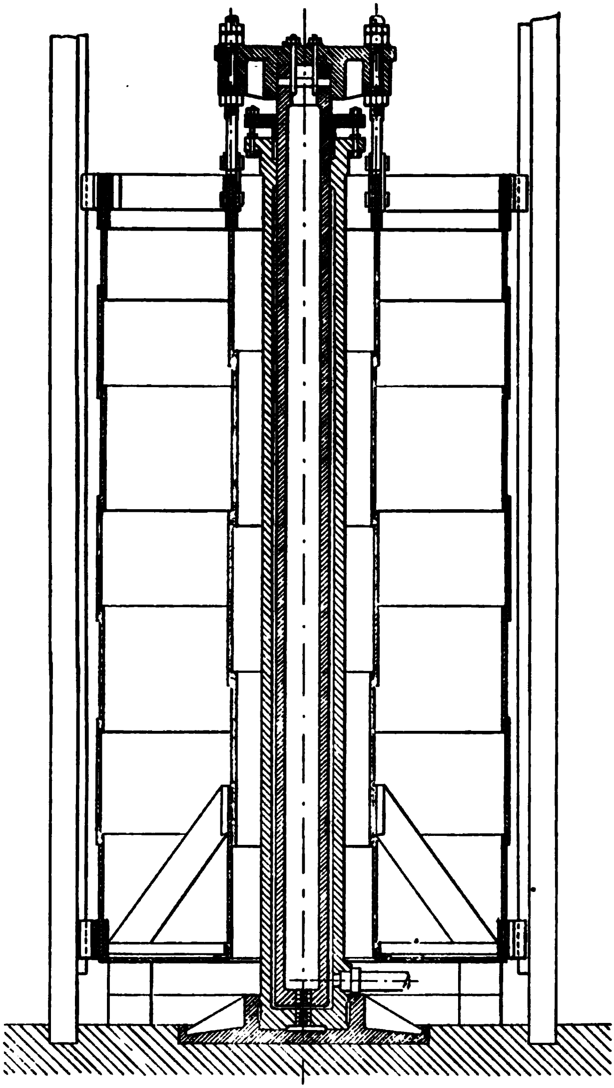


Fig. 109.

or under the influence of an added weight. Thus so long as the accumulator ram is not up to its full stroke the pump will continue to supply water, and will stop when the full stroke is reached.

When the pressure is very slight, and only a small quantity of water is required, a plain ram, as shown in Fig. 107, would not be suitable, on account of the small diameter that would be required. Again, when only a small quantity of water under high pressure is required, a small ram, heavily loaded, might not be possible. In these cases a differential accumulator, as shown at Fig. 108, is employed. These accumulators are used with great success by Mr Tweddell in connection with his hydraulic riveting machines. The ram in the ordinary accumulator (Fig. 107) is free to rise in the cylinder, and carries with it the weight. The cylinder rests in the bottom or base plate, which is securely bolted to the foundations. There is only one gland, and that at the top end of the cylinder. Assuming the ram to be $6\frac{1}{2}$ inches diameter, the area of which is 38.18 inches, and the pressure upon the water to be 700 lbs. per square inch, then the load, together with the weight of the ram, must exceed $33.18 \times 700 = 23,226$ lbs.; whereas, with the differential accumulator, as illustrated in Fig. 108, the same load of 10 tons $7\frac{1}{2}$ cwts. is acting upon an annular area obtained from the difference of the two diameters, viz., $7\frac{1}{2}$ and $6\frac{1}{2}$ inches.

Thus— $7\frac{1}{2}$ in. diameter = 44.17 in. area, and
$6\frac{1}{2}$ „ = 33.18 „

Net area, say = 11.0 in.

$$\text{Pressure per square inch} = \frac{23226}{11} = 2111 \text{ lbs.}$$

Similarly, if only a light pressure of 700 lbs. per square inch is required from the differential accumulator, then the load must include the weight of the moving cylinder, which

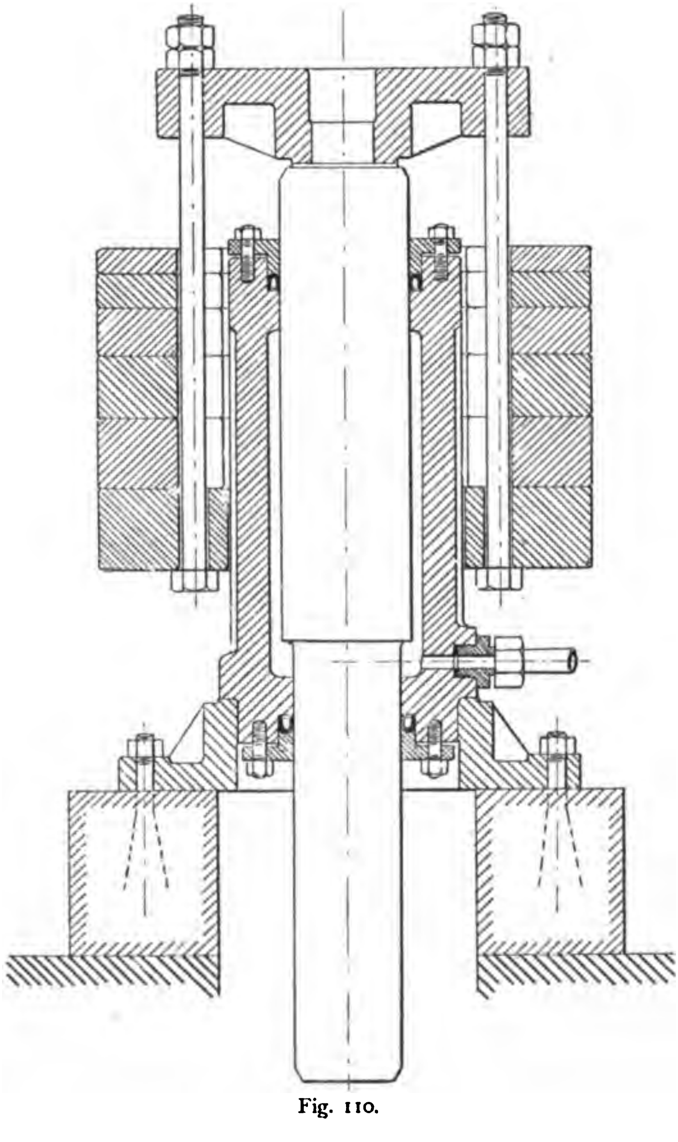


Fig. 110.

has two stuffing glands, one over each part of the ram, as indicated. The weight then upon the column or ring of water within the cylinder will be $700 \times 11 = 7,700$ lbs., as against 23,226 lbs. in the simple accumulator.

The cylinder of the differential accumulator in Fig. 108 is in reality the load plate in addition to the water cylinder. Chocks of timber are provided for the weight to rest upon when right down and not in use. Fig. 110 illustrates a fixed cylinder type of differential accumulator, the moving ram working through two packed glands, and a pit being formed beneath the cylinder for the ram end to work within.

Spring-loaded accumulators have been adopted in some cases, but their range is too narrow to require our giving any attention to their construction.

In hydraulic installations it is frequently desirable to produce a very heavy pressure beyond the ordinary working pressure of the power mains, or beyond the working pressure of the machines, such increased pressure being to give a final squeeze in connection with pressing operations or in connection with riveting plants.

A convenient manner of producing this increased pressure is by means of an intensifier which, in its simplest form, is arranged as a piston working within a cylinder, the piston rod passing through a gland-packed cover, and working in a smaller cylinder carried above the main cylinder. The water from the main is admitted underneath the piston in the large cylinder, and the whole pressure upon it is transmitted by the piston rod or plunger on to the water within the small cylinder, the difference in area of the main piston and the piston rod or plunger giving the difference in pressure between the supply main in the lower cylinder, and the intensified main delivery from the upper cylinder. After the water has been withdrawn from the intensifier cylinder, and used in giving the final pressure, the main cylinder valve is opened to the exhaust, and the water from the intensifier main connection is returned into

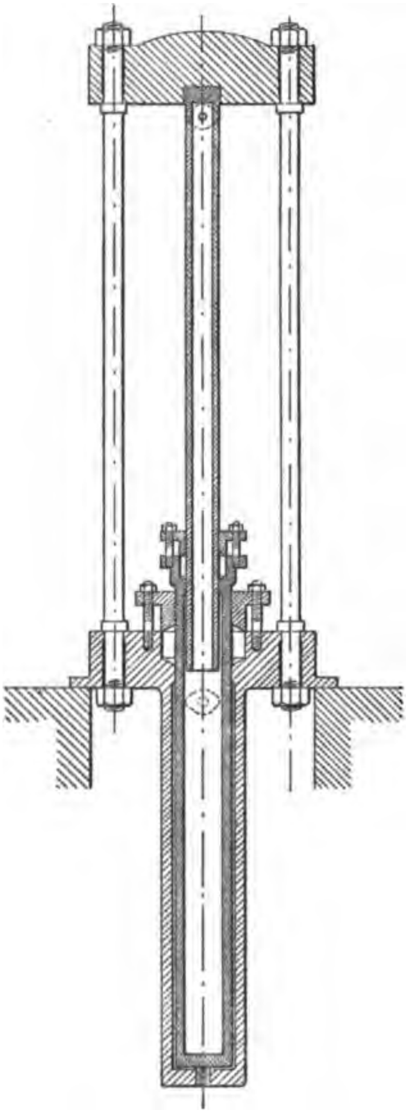


Fig. 111.

the upper cylinder, forcing downwards the main piston in the lower cylinder.

Fig. 111 illustrates an intensifier for use with a water pressure of 750 lbs., the water from the mains entering the lower cylinder, and forcing upwards the hollow ram working upon the upper fixed hollow plunger. The intensified pressure from within the hollow ram and the hollow fixed plunger guide is delivered through the connection shown at the upper end of the fixed ram, while the supply main connection is shown near the base block of the outer cylinder. The ratio of areas of the main ram and the hollow fixed ram or plunger gives the degree of increase of pressure produced. The use of an intensifier of this type in London, where

the main ram was 15½ inches diameter, and the fixed ram 6 inches diameter with a stroke of 13 feet, in a manufactory for making lead pipes, displaced steam of about 15 h.p., and the cost from the public supply mains compared favourably with the old system, notwithstanding the fact that steam power was still in use for other purposes in the same manufactory.

PART VI.—HYDRAULIC PRESSES.

CHAPTER XIII.

PRESSES FOR BALING AND OTHER PURPOSES.

ALTHOUGH the principle of this class of machinery was first stated by Pascal, it was some one hundred and fifty years later ere Bramah usefully applied it to the construction of a press. Pascal's statement has been given in full in Chapter I., and amounts to saying that the pressure on a piston is directly proportional to its area.

Bramah's closed vessel consisted of a pipe having attached to one end a pump barrel, which formed the smaller cylinder, and to the other end a large cylinder containing a ram, and having a cup leather packing, then for the first time used. The large cylinder had four tension bars attached to it which supported a head or table, and the ram carried a similar table or platten. On placing articles on the platten, and operating the pump, a multiplied pressure was given to the article placed on the platten. The modern baling press is a repetition of Bramah's apparatus on an enlarged scale.

The very general use of the hydraulic press, in one form or another, warrants special attention being given to the construction and details of the parts required for particular purposes. Presses are employed for compressing fibrous material, as cotton, wool, esparto grass, peat moss, etc., into small bulk for shipment; for extracting oil and essences from seeds or roots, for embossing paper and printing linoleum, also for sheet metal working and forging operations.

Baling presses are generally provided with a wood or iron box mounted on wheels and having a loose bottom. The

material is packed tight by hand in this box, which is then placed in the press, and the ram pumped out, forcing the loose bottom upwards, and compressing the material. For the greater part of the run out of the ram but little pressure is required, as the material offers only a slight resistance, but after the ram has run out about four-fifths of the height of the box, the pressure rapidly increases owing to the great resistance of the material to further compression.

An inspection of Table X., which gives the pressures in tons per square foot of platten or bottom of baling box to bale cotton, wool, hay, and esparto grass to a given weight per cubic foot, reveals the rapid increase of resistance to compression of these materials after the ram has run out four-fifths of the box.

TABLE X.

PRESSES FOR BALING: PRESSURE IN TONS PER SQ. FT. OF PLATTEN TO BALE MATERIAL TO THE WEIGHTS GIVEN.

Weight in Pounds per Cubic Foot.	Cotton.		Wool, Slightly Greasy.		Hay.		Esparto Grass.	
	Relative Density.	Pressure.	Relative Density.	Pressure.	Relative Density.	Pressure.	Relative Density.	Pressure.
80	20	350	18.82	250	...	...	...	...
70	17.5	180	16.47	140	...	...	...	...
60	15	100	14.11	70	12	60	...	...
50	12.5	50	11.76	35	10	31	10	80
40	10	25	9.41	15	8	14	8	30.3
30	7.5	10	7.05	6	6	5	6	15.5
20	5	3.5	4.7	2.25	4	1.5	4	2.25
15	3.75	1.8	3.5	1.15	3	.67	3	.9
10	2.5	1.1	2.37	.6	2	.3	2	.3
Weight in Pounds per Cubic Foot, Hand-packed.	4	...	4.25	...	5	...	5	...

The baling box should be $1\frac{1}{4}$ inches less in length, breadth, and height than the size of bale required. The

pressures in the Table are for the compression only, and an allowance for the friction of the material against the sides of the baling box must be added. For bales of 40 lbs. per cubic foot and under add 25 per cent. to the above pressures, and for heavier bales add 40 per cent.

Large baling presses are usually supplied with hydraulic pressure pumps driven by steam power, and, as the available power is constant, while the work to be performed varies greatly, many arrangements have been tried for saving time, although the one usually adopted consists of a battery of pumps arranged in groups. The pumps are all set to work during the earlier part of the stroke, thus driving out the ram at a rapid rate. When the pressure rises, so that the work done by the pumps is the maximum available from the steam plant, one group of pumps is automatically tripped or put out of action in a manner to be described in Chapter XVI. The remaining pumps continue to work until the further rise of pressure causes the power to reach the maximum, when another group of pumps is tripped. This tripping is continued until the last group of pumps only remain, and these are so proportioned that they trip when the bale is of the required density. By properly proportioning the pumps the ram can be driven out in the shortest space of time possible for any number of pump plungers and power available. We will illustrate this fact by first considering the case where only two pump plungers are used.

In the case of a pump having two or any other number of plungers, the smallest plunger is fixed in size by causing it to absorb the maximum power available when on the point of tripping. The remaining plunger may be given any size, and must be arranged to trip out when such a pressure is reached that the two plungers working together absorb the maximum power available. There is, however, a size for this larger plunger, which will cause the ram to travel out in the shortest time. As the equations to the curves of pressures for the different materials are unknown, it is im-

possible to give an equation for finding the size of the larger plunger; the graphic method in Fig. 112, however, gives very close approximations to the truth. Fig. 112 represents the curve of pressures for baling hay to a weight of 50 lbs. per cubic foot, or to a bulk of one-tenth that of hand-packed hay. A B represents the length of the baling box filled with

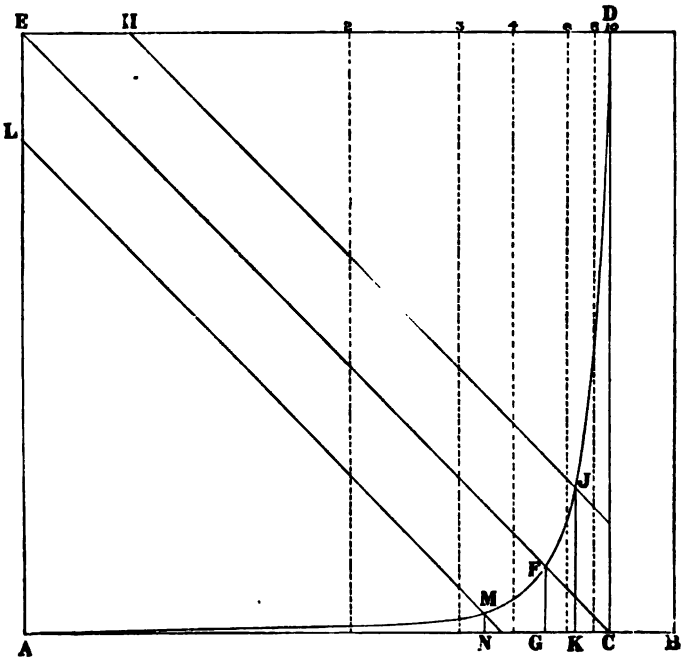


Fig. 112.

hay, A C the stroke of the ram, and C B the final depth of the bale of hay. The curve A M F J D is the curve of pressures per square inch of pumps drawn out to a scale making C D equal to A C. This curve is ascertainable from Table X. Complete the square A C D E, and join E C, cutting the curve in F, and draw the vertical F G, then F G represents

the pressure per square inch at which the larger pump must trip. This pressure being known, the combined area of the two pump plungers can be fixed such that the total available power is absorbed at this pressure. The size of the smaller plunger being already fixed, the larger is ascertained by subtraction.

If three pump plungers are to be used, the pressures at which the two larger must trip can be found by drawing the diagonals L M, H J so that the area of the square A C D E is divided into three equal parts, or

$$AC^2 = 3\frac{AL^2}{2},$$

from which A L may be found. The diagonals L M, H J having been drawn, the verticals M N, J K give the pressures at which the pumps must trip. The sizes of the plungers are now ascertained by finding the combined area of the small and medium plungers, at the pressure J K, to absorb the maximum available power, from which area the size of the medium plunger is found, as before, by subtraction. In the same way the combined area of the three plungers is found for the pressure M N, and the size of the largest ascertained by subtracting the combined area of the other two.

An example will render the process more clear. Hay is to be baled to a weight of 50 lbs. per cubic foot, and the press is to be worked with a maximum pressure of 2 tons per square inch. The maximum available power is 3¾ horse-power. Referring to Table X., the weight of hand-packed hay is 5 lbs. per cubic foot. When compressed to 50 lbs. per foot, the space occupied will be one-tenth, or the ram must travel nine-tenths up the baling box. A B and A C can now be laid down, making C B one-tenth of A B. Construct the square A C D E, and draw the curve of pressures, making C D represent 2 tons. Draw the diagonal E C,

and scale off F G, which in this case is .217 ton. The sizes of the plungers may now be settled.

$$3\tfrac{3}{4}\text{ h.p.} = 33000 \times 3.75 = 123{,}750 \text{ foot-lbs. per minute.}$$

Efficiency of pumps, say .66.

$$\text{Energy available} = 123750 \times .66 = 82500 \text{ foot-lbs. per minute.}$$

The velocity of the plungers may be anything up to 50 feet per minute. In the case under consideration the small plunger may be made 1 inch diameter, and its travel in feet per minute will then be

$$\frac{82500}{.7854 \times 2 \times 2240} = 23.16 \text{ feet.}$$

A 1-inch plunger working against a pressure of 2 tons per square inch requires to travel through 23.16 feet to develop 82,500 foot-pounds of energy. As the plungers are only single acting, the actual velocity of the plunger becomes $23.16 \times 2 = 46.32$ feet per minute, which is under 50 feet velocity. The stroke and consequent number of revolutions may be settled last.

The size of the larger plunger may now be ascertained. Let A be the area in inches of the larger plunger—

$$(.7854 + A).217 \times 2240 \times 23.16 = 82500 \text{ foot-lbs.}$$

$$A = 6.68 \text{ square inches.}$$

$$3 \text{ inches diameter} = 7.07 \text{ inches area.}$$

Therefore we may use a 3 inches diameter plunger tripping at a pressure of 450 lbs. per square inch, and a 1 inch diameter plunger tripping at a pressure of 2 tons.

By adopting a stroke of 4 inches, the number of revolutions per minute of pump shaft is

$$23.16 \times 3 = 70.08 \text{ revolutions.}$$

The large ram of the press must be proportioned to give a pressure of 31 tons per square foot of platten, with an addition of 40 per cent. to overcome friction of baling box,

and 2 tons for stuffing box friction, making a total of 45 tons per square foot of platten.

If three plungers had been desired, the smallest would still be the same size—1 inch diameter. The two larger ones are found by drawing the two lines L M, H J in Fig. 112, as directed, and scaling M N, J K.

$$\begin{aligned} JK &= .48 \text{ tons per square inch.} \\ MN &= .062 \quad\text{,,}\qquad\text{,,} \end{aligned}$$

The area A of the middle plunger is found as before, but for a pressure of .48 tons.

$$(.7854 + A).48 \times 2240 \times 23.16 = 82500.$$
$$A = 2.53 \text{ square inches.}$$
$$1\tfrac{3}{4} \text{ inches diameter} = 2.40 \text{ inches area.}$$

The area B of the large plunger may now be found—

$$(B + .7854 + 2.40).062 \times 2240 \times 23.16 = 82500.$$
$$B = 22.47 \text{ square inches.}$$
$$5\tfrac{1}{4} \text{ inches diameter} = 21.64 \text{ inches area.}$$

The plungers to be used are $5\frac{1}{4}$ inches diameter tripping at 132 lbs. per square inch, $1\frac{3}{4}$ inches diameter tripping at 1,085 lbs. per square inch, and 1 inch diameter tripping at 2 tons per square inch.

If in designing the $5\frac{1}{4}$-inch plunger gives trouble and requires a wider spacing of the cranks than is necessary for the strength of the crankshaft, the stroke may be increased to 6 inches, and the diameters of plungers reduced accordingly.

4 inches increased to 6 inches.

1	in. diam.	$=.7854$	area reduced	$=.7854 \times \frac{4}{6} = .52$	in. area	$= \frac{13}{16}$	in. diam.
$1\frac{3}{4}$	,,	$=2.40$	,,	$=2.40 \times \frac{4}{6} = 1.6$	,,	$= 1\frac{1}{2}$	,,
$5\frac{1}{4}$	,,	$=21.64$	,,	$=21.64 \times \frac{4}{6} = 14.42$	,,	$= 4\frac{1}{4}$	,,

Fig. 113 represents the usual form of hydraulic press, having a cylinder A of cast iron or steel, the latter being much in request for presses for export, as the weight is then only

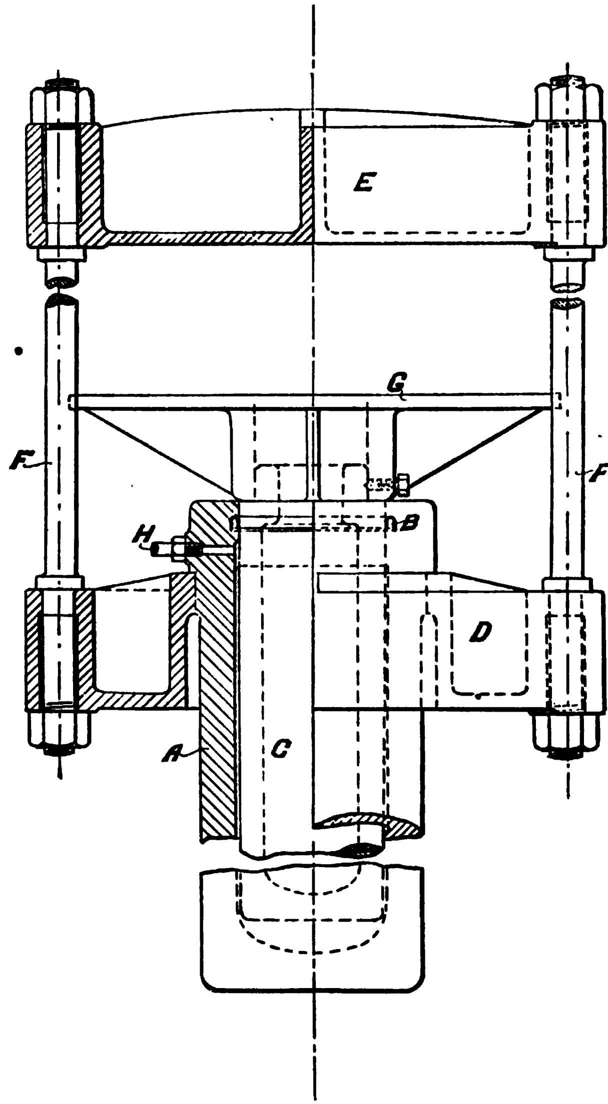


Fig. 113.

about one-third. The cylinder has a U leather packing B and ram C, and rests on the faced edge of the base plate D. The head E of the press is attached to the base D by bolts, or pillars F, usually four in number. The ram carries a platten G, on which the article to be pressed rests. Water is admitted to the cylinder at H. Instead of the pillars F being made, as shown, with a nut at each end, they are sometimes made with two forged collars at each end, and the bosses of the head and base are split to receive them, and fitted with caps bolted on. The platten is guided by

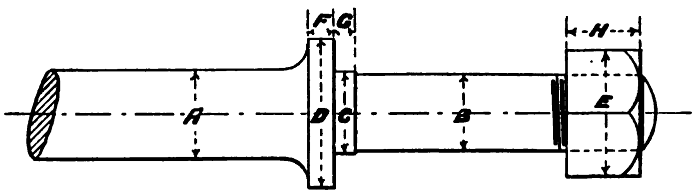


Fig. 114.

the bars F, and has its corners curved to fit the bars, or sometimes slipper guides are bolted to the platten to increase the rubbing surface.

Suitable thicknesses for the cast-iron rams are given in the following table :—

Diam. of Ram in Inches.	6	8	10	12	14	16	18	20
Thickness of Ram in Inches.	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{8}$	$2\frac{1}{4}$	$2\frac{3}{8}$	$2\frac{1}{2}$

The pillar F is shown in detail in Fig. 114, and the sizes are given in Table XI. (next page).

TABLE XI.

SIZES OF WROUGHT-IRON BARS FOR PRESSES.

4 Bars to a Press.

Press Test Load.	A For length of 1′ to 4′	4′ to 7′	7′ to 10′	10′ to 13′	B.	C.	D.	E.	F.	G.	H.
Tons.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.
10	1¼	1½	1⅝	...	1¼	1⅜	2½	2	¾	⅝	1¼
20	1¼	1½	1¾	...	1¼	1⅜	2½	2	¾	¾	1¼
30	1⅜	1½	1¾	...	1⅜	1½	2½	2 3/16	¾	¾	1⅜
40	1½	1¾	2	2¼	1½	1⅝	2⅞	2⅜	¾	⅝	1½
60	1⅞	2	2¼	2½	1⅞	2	3⅛	3 1/16	⅞	⅞	1⅞
80	2⅛	2¼	2½	2¾	2⅛	2¼	3¼	3 5/16	⅞	⅞	2⅛
100	2⅜	2½	2⅝	3	2⅜	2½	3⅜	3¾	⅞	⅞	2⅜
150	2¾	3	3¼	3½	2¾	2⅞	3⅞	4 3/16	1	1	2¾
200	3¼	3¼	3¼	3½	3¼	3⅜	4	4⅞	1	1	3¼
300	4	4	4	4¼	4	4¼	5⅜	6	1	1	4

Fig. 115 is a plan of the usual form of press-head. The stresses occurring in the head vary according to the manner in which the load is distributed, and are worthy of investigation.

In any manner of loading in which the centre of pressure of the load coincides with the centre of the head, or at the intersection of G H, N O, the load is equally distributed to the four pillars at A B C D, and if W represents the total load or pressure on the head, each pillar carries a load of $\frac{W}{4}$. Whatever share of the load is carried by the ribs EF, GH, IK, and LM, NO, PQ is transmitted to the four side ribs AD, BC, and AB, CD, which in turn transmit the load to the pillars. As the four ribs AD, BC, AB, CD, each carry an equal load, that load is evidently $\frac{W}{4}$.

Four typical methods of loading have been selected for investigation, and any others may be considered as similar to one or other of these with sufficient accuracy.

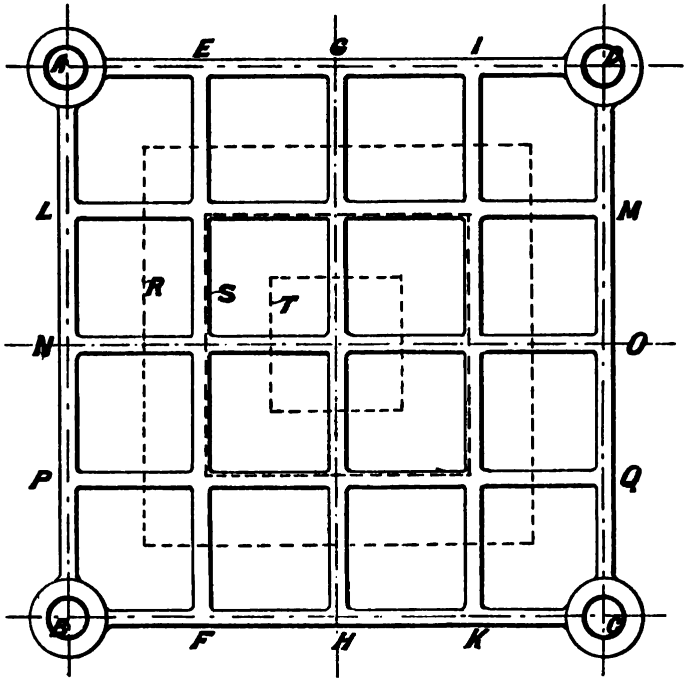


Fig. 115.

These four methods of loading are :—

(1.) Load distributed over the whole press head within the centre lines A B C D.

(2.) Distributed over the area bounded by the line R.

(3.) „ „ „ „ S.

(4.) „ „ „ „ T.

In all these cases the load W may be divided into two parts $\frac{W}{2}$; one of which is supported by the ribs running in one direction, as AB, EF, GH, IK, DC; and the other by the ribs at right angles to these, as AD, LM, NO, PQ, BC.

In (1) the load $\frac{W}{2}$ carried by AB, EF, GH, IK, DC is divided up as follows:—

$$\frac{W}{16}+\frac{W}{8}+\frac{W}{8}+\frac{W}{8}+\frac{W}{16}=\frac{W}{2},$$

and a similar load is carried by the remaining bars, so that AD carries a load of $\frac{W}{16}$.

AD also carries half the loads of EF, GH, IK, so that the total load on AD is—

$$\frac{W}{16}+\frac{W}{16}+\frac{W}{16}+\frac{W}{16}=\frac{W}{4} \text{ as above stated.}$$

The bending moments may now be expressed:—

$$\text{for AD}=5\frac{WL}{128},$$

$$\text{for EF, GH, or IK}=\frac{W}{64}.$$

The load on AD is not evenly distributed.

In (2) the load $\frac{W}{2}$ is carried by EF, GH, IK as follows:—

$$\frac{W}{6}+\frac{W}{6}+\frac{W}{6}=\frac{W}{2}.$$

Half of these loads are carried by AD, or—

$$\frac{W}{12}+\frac{W}{12}+\frac{W}{12}=\frac{W}{4}.$$

The bending moments in this case are:—

$$\text{for AD} = \frac{WL}{24},$$

$$\text{for EF, GH, or IK} = \frac{WL}{48}.$$

In (3) the load $\frac{W}{2}$ is carried by EF, GH, IK, as follows:—

$$\frac{W}{8} + \frac{W}{4} + \frac{W}{8} = \frac{W}{2}.$$

Half of these are carried by AD, or—

$$\frac{W}{16} + \frac{W}{8} + \frac{W}{16} = \frac{W}{4}.$$

The bending moments are:—

$$\text{for AD} = \frac{3WL}{64},$$

$$\text{for EF or IK} = \frac{WL}{64},$$

$$\text{for GH} = \frac{WL}{32}.$$

In (4) the load $\frac{W}{2}$ is carried by GH—

$$\frac{W}{2} = \frac{W}{2}.$$

Half of this is carried by AD—

$$\frac{W}{4} = \frac{W}{4}.$$

The bending moments are:—

$$\text{for AD} = \frac{WL}{16},$$

$$\text{for GH} = \frac{WL}{8},$$

$$\text{for EF or IK} = O$$

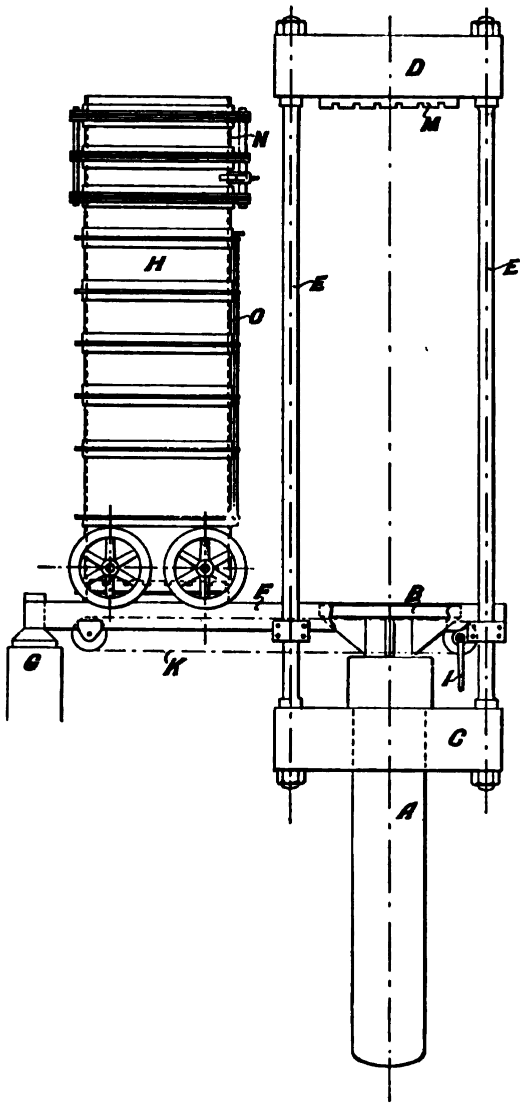


Fig. 116.

The section of metal required may now be determined. The flat plate of metal forming the face of the head is made of the same thickness of metal as the ribs, and may be included in taking out the sizes.

By first neglecting the flat plate each rib may be determined as a rectangular section by equating its bending moment to the moment of resistance of a rectangle. Thus for AD in (1)—

$$\frac{5WL}{128} = \frac{bh^2f}{6}$$

where b is the width of the rib, h the height, and f the stress to which the metal is to be subjected. If L is in feet, b and h must also be in feet, and if w is in tons, f must be in tons. By selecting values for b and f and solving this equation, the value of h can be found; it is usual to fix values for b and f, and if h is unsuited, b must be varied and another value found for h.

These values being settled, the correct height h_1 of the head can be ascertained as follows:—

Let r be the distance between the ribs.

$$\text{Then } h_1 = \frac{h}{2} + \frac{h^2}{4r} + \frac{b}{2}.$$

To avoid difficulties in casting it is usual to find the dimensions of the strongest rib, and make the others of the same dimensions.

The formulæ have been worked out for square heads with three intermediate ribs. They are, however, applicable to any rectangular head with not less than three intermediate ribs, and having the load distributed over a rectangle having the same ratio of sides as the head.

Fig. 116 represents a baling press and box complete. The cylinder A, containing the ram carrying the platten B, is carried by the base C, which in turn is connected to the head D by the bars E. Guide rails F are attached to the bars E and supported at G. The baling box H runs on grooved

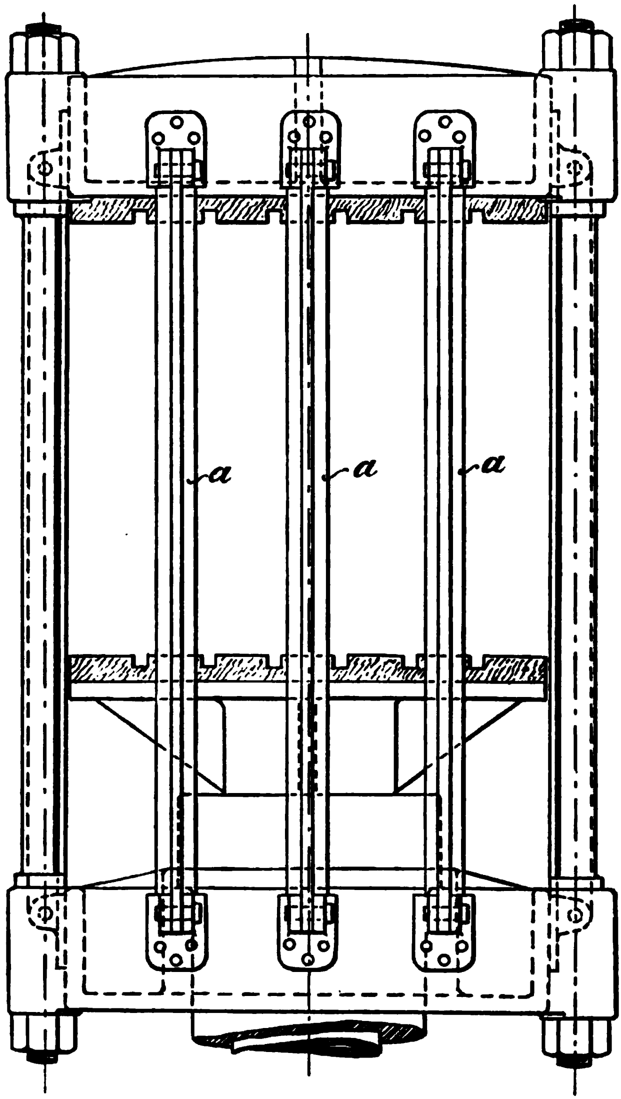


Fig. 117.

wheels resting on the rails F. The bottom of the baling box consists of a piece of grooved hardwood resting on a ledge or fillet. The cotton or other material to be baled is packed in the box H by hand, and the box is then drawn into the press by revolving the handle I which causes the chain K attached to the baling box H to travel, and so move the box H. The front of the box H is cut away at the top to clear the hardwood block M on the press-head. When the box is central over the ram, the water is pumped into the cylinder A, and the platten B passed up inside the box H carrying the hardwood bottom with it.

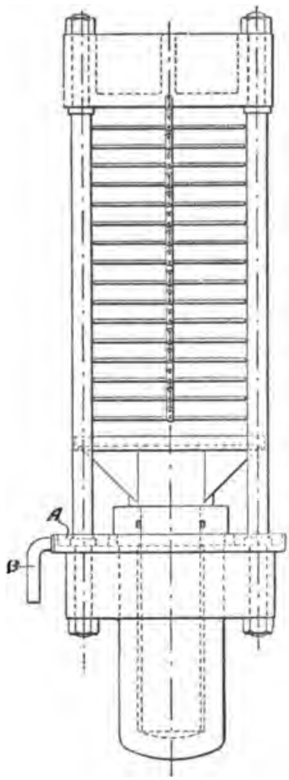


Fig. 118.

When the baling operation is completed, the hinged doors N and O of the baling box are opened, and the box H withdrawn by turning the handle I. The upper part of the baling box H has three of its sides hinging outwards to allow of the expansion of the bale on the pressure being released.

The box H having been removed from the press, the doors N and O are closed and refilling commenced. The bale in the press is at the same time hooped with iron bands which are passed through the slots in the hardwood blocks, and secured round the bale by riveting or other suitable means. The ram is now lowered into the cylinder, and the bale removed.

Fig. 117 represents a dumping press which is made in exactly the same way as the hydraulic presses already noticed, but in place of the baling box it is supplied with steel bars *a*, *a*, *a*, of strong T section. The bars are usually hinged at the base, and fitted with a draw-pin at the head of the press. The material is lightly baled in up-country districts in screw-power presses, and when brought to the quays is pressed or dumped to the requisite size for shipment in a press of this description.

Fig. 118 represents one form of hydraulic oil-press suitable for extracting oils or essences from seed and roots. The press is precisely similar to a baling press in the main features, but has a series of hanging plates or platforms equally spaced as shown. The seeds or roots are placed in flat canvas or horse-hair bags, which are placed on the plates, and the press is then operated. The oil or essence escaping trickles off the edges of the plates, and over the down-turned edge of the platten into the flat tank A, where it is run off by the pipe B into suitable vessels.

CHAPTER XIV.

SHEET METAL WORKING AND FORGING MACHINERY.

It was about the year 1860 that the hydraulic press was first practically used for the forging of ingots for big guns at Messrs Whitworth's. About the same time the English engineer in charge of the Vienna locomotive shops introduced a hydraulic press for forming the various details of locomotives and railway stock. The work done was of a varied nature, including forging in closed dies, punching, and drawing out and dumping operations.

Needless to say, there was the usual prejudice to the new tools and methods, but their superiority was evident to leading engineers, so that at the present time the hydraulic press is almost solely used for large work, whilst its popularity for small work is rapidly increasing.

Before passing to the hydraulic machine tools proper we will notice a small hand-worked punching bear illustrated in Fig. 119. The punch is attached to a ram A fitted with a cup leather, and working in a cylinder B formed in the main frame of the bear. The cylinder is surmounted by a water cistern C containing a pressure pump worked by a hand lever D. When the pump is worked water is forced into the cylinder B, so driving the ram down and forcing the punch through the metal. To raise the punch clear of the work the thumbscrew E is loosened, and the cam F attached to the lever G is operated, thus driving the water back into the cistern C.

We will now consider the usual types of forging presses in

use. Fig. 120 gives a general idea of the arrangement of a large forging press. The cylinder A, carrying the ram B, is supported by two or four vertical columns C, secured to the base D, which carries the anvil or bottom die. Two cylinders E F are fixed to the press, and are always open to the pressure water, so that when the large cylinder A is open to exhaust, the pressure acting on the rams G H drives the ram

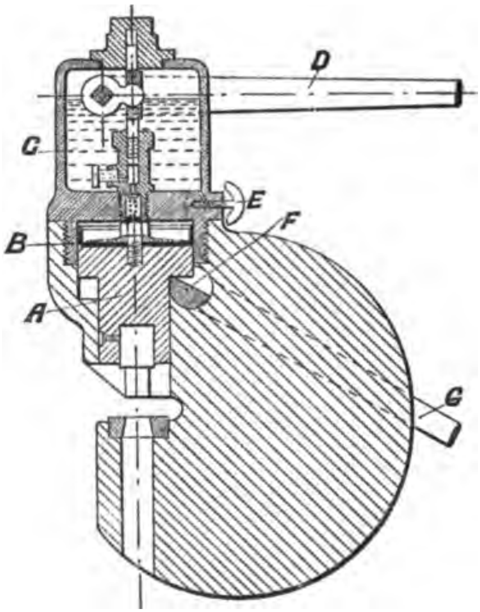


Fig. 119.

B up, thus the press is controlled by one valve only. In many designs of press the drawback cylinders E F are placed the reverse way up, and are secured to the large head casting, being then provided with tension rods to lift the ram B. Arrangement of minor points must, however, be governed by circumstances, as if the press is too lofty it will interfere probably with the passage of overhead travelling cranes.

When four columns are used, they are so disposed as to keep the press as narrow as convenient in one direction, so that the tackle for handling the forging may be brought as close as possible to the dies. Various methods are adopted for securing the head to the columns; in the press shown the head rests on a collar or neck formed on the column, and is secured by a nut. In another method the column has two collars formed near each end, and the head and base castings have bosses bored out and fitted with caps; the column is inserted and the cap bolted on.

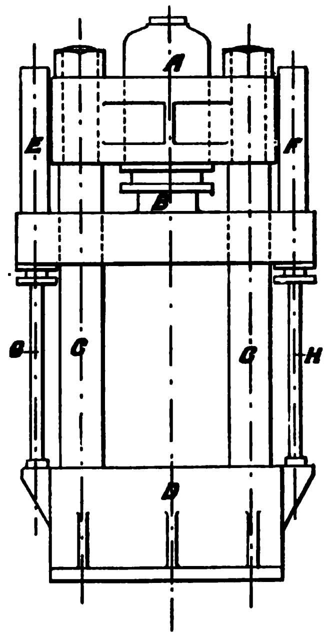


Fig. 120.

In some presses provision is made for altering the depth of the gap or space between the ram and anvil, or "daylight," as it is called. This is generally done by placing the cylinder at the bottom, and the top casting is made adjustable by having the pillars screwed for a considerable length, and provided with two nuts for locking the casting in the required position.

Different firms have at times produced presses varying in design and claiming special advantages. The Davy press has two cylinders placed side by side, and attached to one common crosshead, the crosshead being provided with a long arm projecting upwards from its centre, and having a turned cylindrical part at its upper end. This cylindrical part works

in a tubular guide placed between the two cylinders, and together with the guides working on the columns forms a triangular support, giving great steadiness to the top die. Another advantage of this form of press is that the pressure on the dies may be considerably off the centre line of the press without causing severe straining.

A multiple power press, designed by Messrs Tweddell, Fielding, & Platt, has three equal-sized rams placed side by side below the floor level, the rams all acting on a common crosshead, connected by strong tension bars, which also act as guides to the head of the press carrying the top die; whilst the bottom die is supported by a base which also carries the three hydraulic cylinders. Three different powers are obtained by this arrangement, according to the number of rams acted upon by the pressure water. This press also has the advantage that the head room is unobstructed, thus allowing a free passage for travelling cranes.

For very large forging presses it is not usual to work with an accumulator, the water being supplied direct from a pump into the cylinder, the idle part of the stroke being performed by the pressure of water contained in an overhead tank. The lifting cylinders are frequently operated by a steam accumulator, or by pressure water from an ordinary accumulator.

By another method the pressure is applied by a direct steam driver, which consists of a large steam cylinder coupled direct to a plunger, which is connected without the interposition of valves to the press cylinder, the steam cylinder being operated by an ordinary slide valve.

It is absolutely necessary to use a high pressure in the cylinders—usually 2 to 3 tons per square inch—otherwise the machines become very costly and heavy in weight, or the manufacture is rendered impossible.

Fig. 121 shows the usual arrangement of a small open-sided or C press, which can be conveniently made for pressures up to about 150 tons. In the illustration a vertical forging ram

A is shown, also a horizontal ram B, each supplied with a drawback ram constantly open to the water pressure. Two valve levers are shown, one to each cylinder. Hydraulic push-back cylinders are supplied to each ram, and are always subjected to the pressure water. Machines of this type may be used for all kinds of stamping and punching as well as general forging.

Fig. 122 shows a section of the cylinders and ram of a Tweddell punch. The ram A carrying the punch is formed of two circular parts placed eccentric to each other, thus placing the punch well forward and easily visible. The ram A is packed by a U leather, and works in the gun-metal lined cylinder B. The return motion of the ram is effected by the drawback ram or piston C working in the cylinder D, which is always open to the pressure water. A water-saving appliance is added, which is operated by the lever E, and closes the valve when the punch has penetrated the metal.

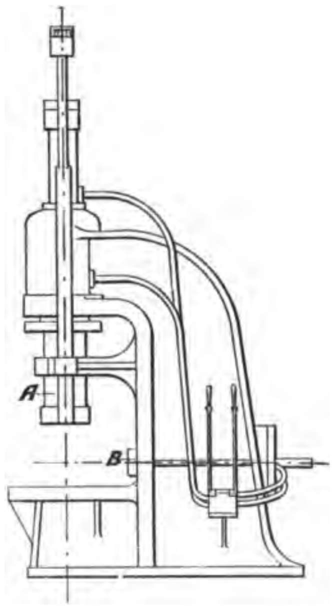


Fig. 121.

The working of the water-saving appliance will be better understood by an examination of Fig. 123, which illustrates a manhole punch or flanging press. The only difference between this and the last press lies in the fact that the dies are arranged centrally with the ram. The down stroke of the ram causes an oscillation of the lever F, which by means

of the adjustable tappets or nuts G causes a movement of the hand lever H, which operates the balanced valve I, cutting off the water pressure. The attendant now gives the valve a

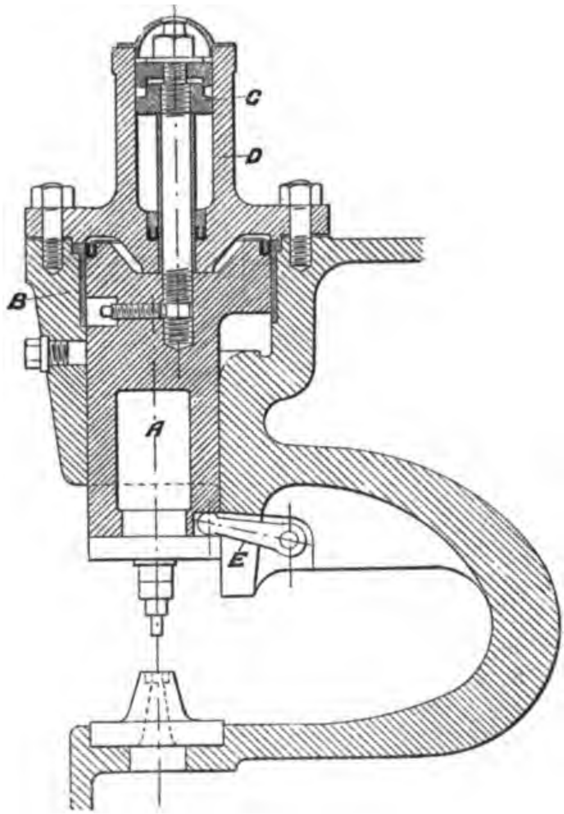


Fig. 122.

further movement, opening it to exhaust, the ram rises, and in doing so oscillates the lever F in the opposite direction, causing the adjustable nuts K to move the hand lever H

and valve I back to the central position, ready to be again operated by the attendant to cause the next down stroke.

Fig. 124 shows a shearing machine having the water-saving mechanism so arranged that the valve may be worked by either hand or foot power. In this machine the drawback ram is placed behind the large cylinder in the main casting.

Fig. 125 illustrates Tweddell's plate-bender, for forming

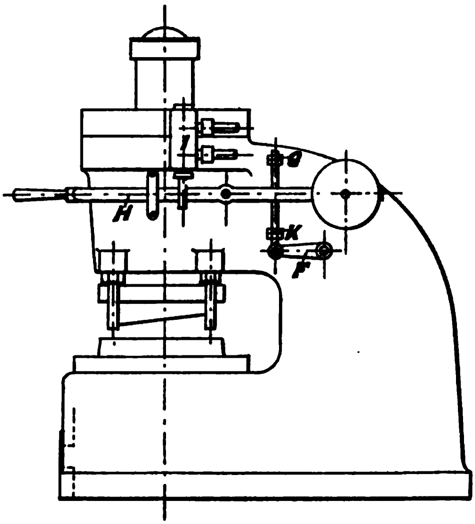


Fig. 123.

the shells of large boilers and for similar work. The plate to be bent is fed through the slot A, water pressure is then applied to the cylinder B, causing the die C to advance towards the die D, so bending the plate. The die C is returned, on the cylinder B being opened to exhaust, by the drawback ram E. The plate is now further advanced and another stroke of the die C given. By this means the plate is bent to the final curve at the rate of 2 to 3 feet per

minute. An adjustable stop is provided which prevents the dies coming too close together, and so forming a circle of less radius than is required. The dies are not made to any radius, but the die D has a central rib, while the die C has two ribs a short distance apart. To remove the work the head F is slewed round. A hinged tie-bolt is, however, some-

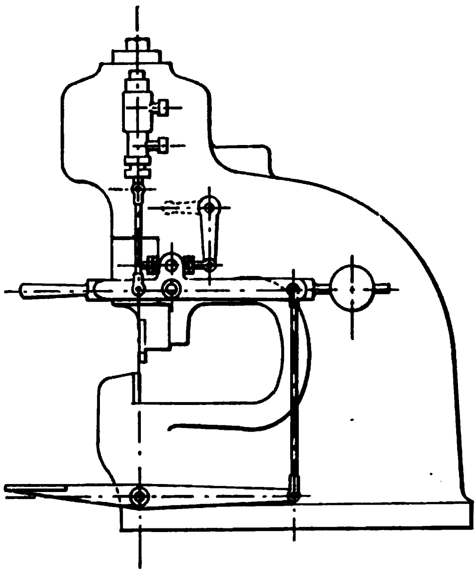


Fig. 124.

times provided to connect the die D to the main frame at its upper end.

Fig. 126 shows in plan the general arrangement of a tube-drawing machine. A hydraulic cylinder A is provided, having connected to it two long bars B supported on feet. The tube to be drawn is first slightly reduced at one end, and then, having been threaded on to the mandrel, is placed in the

machine with its reduced end passing through the die carried in the holder C. The tail end of the mandrel is now attached to the support D, and the reduced end of the tube is gripped by the jaws E carried by the crosshead F, capable of being drawn along by the water pressure acting on the piston G. The stroke being completed, the tube is removed and the crosshead F returned by water acting on the back of the

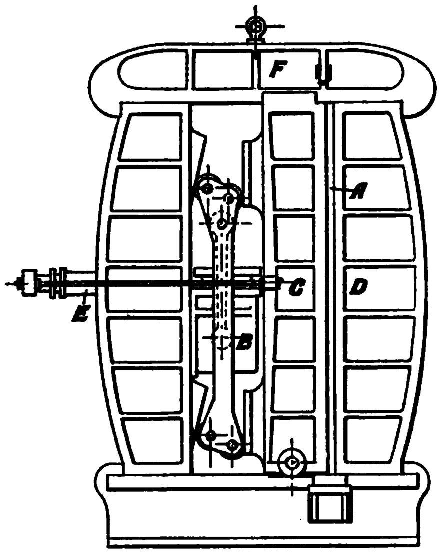


Fig. 125.

piston G, the water in front being returned to the accumulator.

Fig. 127 shows a hydraulic press arranged for putting on and taking off railway rolling stock wheels. The action will be readily understood from the illustration. The wheels to be operated upon having been suitably adjusted between the tension bars A, the ram B is pumped out by the hand pump C, so forcing on or taking off the wheel.

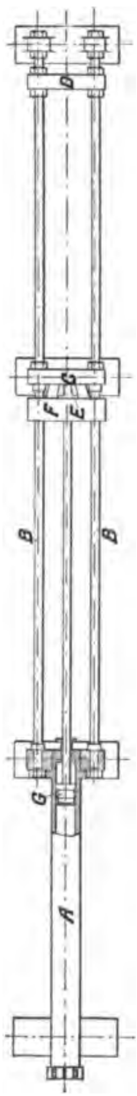


Fig. 126.

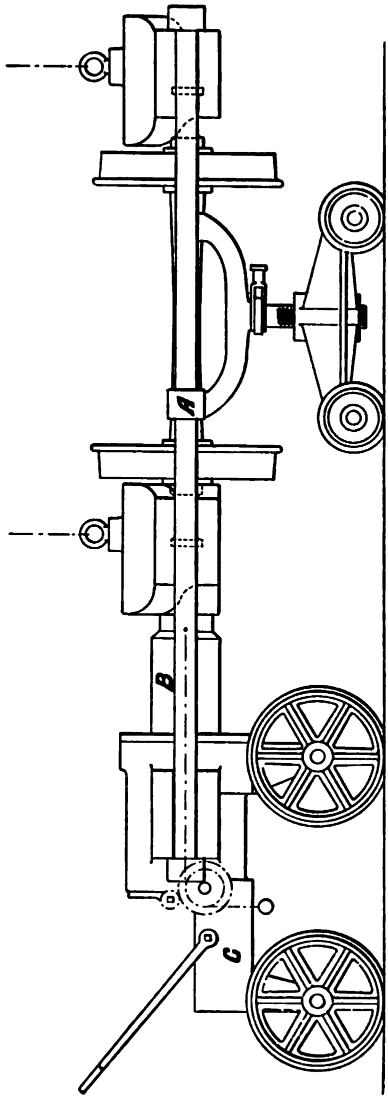


Fig. 127.

CHAPTER XV.

HYDRAULIC RIVETERS.

THE very general use of the hydraulic riveter for shipbuilding, boiler-making, and girder work is undoubtedly due to the efforts of the late Mr R. H. Tweddell and to Messrs Fielding & Platt.

Water power is particularly suitable for riveting, in that the machines consume no energy except when actually at work, and being portable in most cases, can be readily carried to any desired position upon a scaffold or within a structure or frame where ordinary machines having rotating power or shafting could not be employed. The system of laying the hydraulic pipes or mains with union branches at positions likely to be suitable for any special tool, or any part of a building yard or dock, provides, with the use of travelling on telescopic joints, an easily controlled and economical method of mechanical riveting. In Fig. 128 is shown in sectional elevation the motive power end of a portable riveter.

The water pressure is used to give three distinct movements to the operating members. First, in the cylinder formed within the ram B, the ram D being forced out, carrying with it the plate-closing die E, also the ram B and rivet-closing die C. Second, in the cylinder A forcing out the riveting die C, thus closing the rivet. Third, the water always at constant pressure on the ram H, within the draw-back cylinder G, carries back or returns the rams B and D when the other cylinders are opened to exhaust. The water from L enters the cylinder within B by means of the sliding

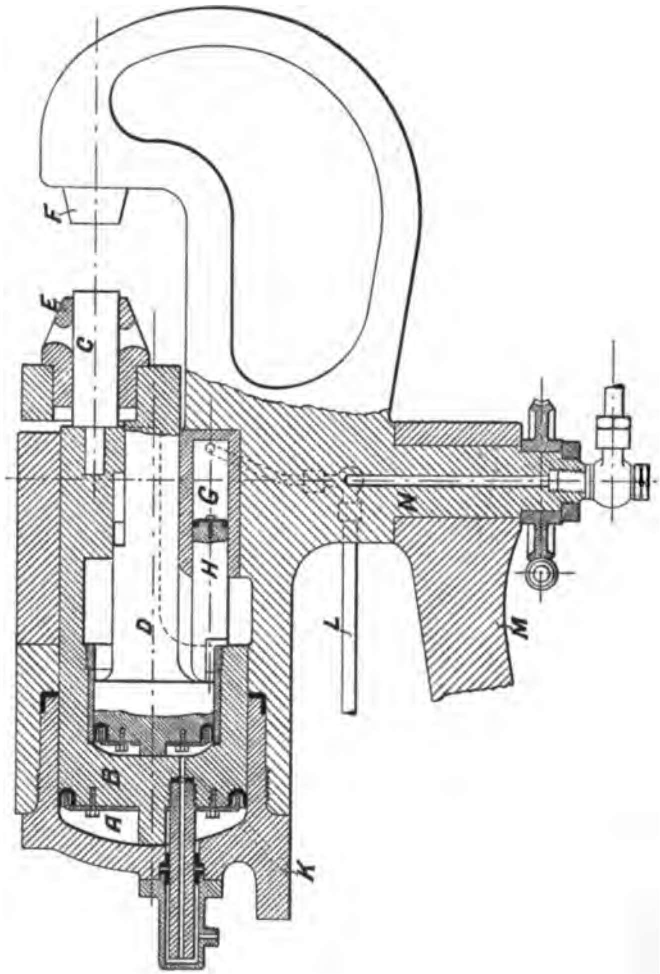


Fig. 128.

packed joint at the upper end of the ram B, and the passage shown in dotted lines K admits the pressure from the valve which is usually attached adjacent to the passage. Swivelling is provided for by the suspension arm M being formed with a boss for the frame stem N to pass through.

Fig. 129 illustrates a portable riveter having a hand worm and wheel for swivelling the frame into any position to suit

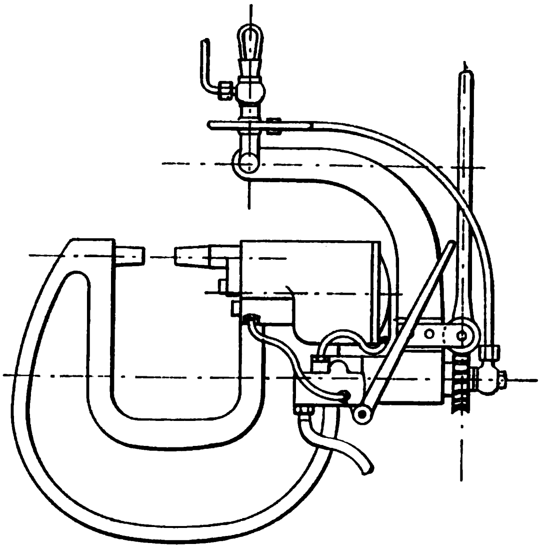


Fig. 129.

the work. The hanger is made of cast steel, and by means of a flexible pipe the necessary movement is obtained without difficulty.

Variable power is sometimes desirable in connection with fixed or portable machines, so as to obtain the best results without necessitating a constant consumption of water when the duty is not a constant one. Fig. 130 illustrates in elevation a double power and plate-closing riveter of 150

tons power, and having a gap of 8 feet suitable for marine boiler plate riveting.

Fig. 131 illustrates in sectional view the motive power end of a plate-closing riveter, having arranged thereon also water-saving rams whereby an economy of about 60 per cent. is obtained. Three valve levers are employed to control the

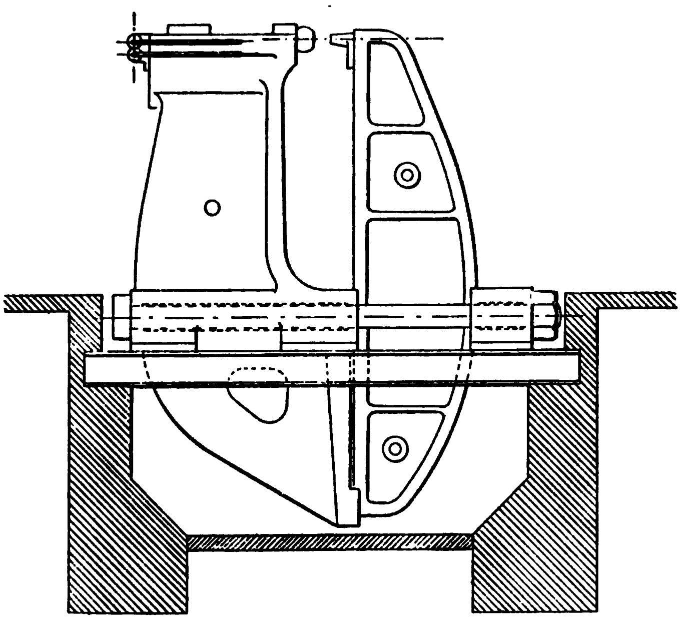


Fig. 130.

three valves A B C, being the water-saving valve, the plate-closing valve, and the main ram valve respectively. Water from a tank having a head of about 20 feet supplies the valve A, which is used to advance the plate-closing ram D, which carries the closing tool *d*. The water-saving and drawback piston ram E is fed by water on both sides of the piston, the difference of area of the full outer against the inner annular

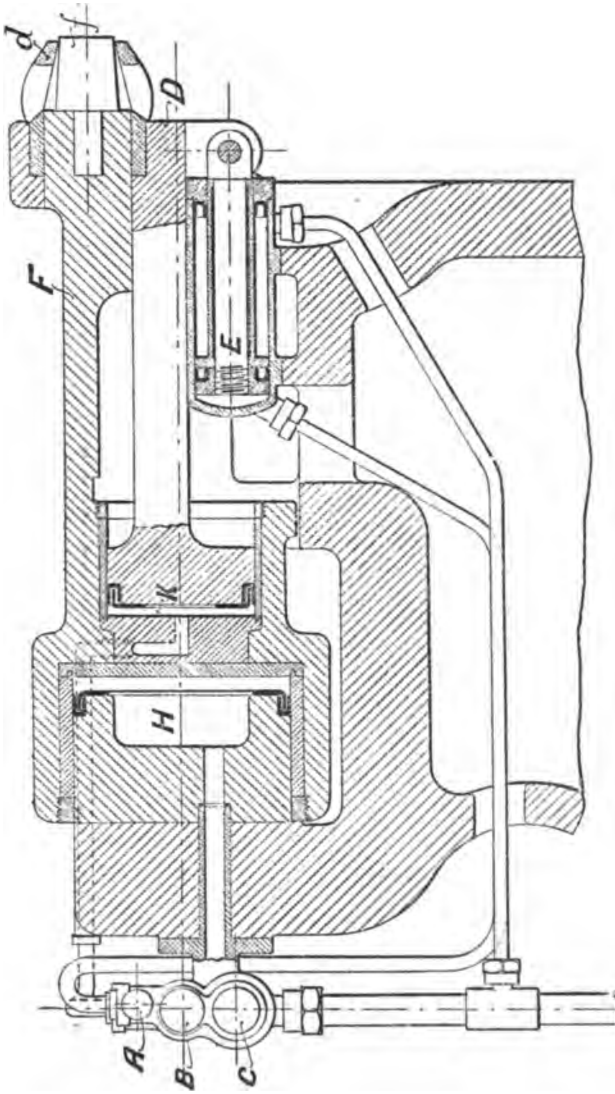


Fig. 131.

end causing the ram E to advance and with it the rams D and F, so that the plate-closing tool *d* and the cupping die *f* are brought close up to the work, the movement being assisted also by the low-pressure or tank water being at the same time taken into the main cylinder H and the plate-closing cylinder K. Pressure water is then admitted into the main cylinder, the effective area being the difference of the areas of the plate-closing ram K and the main cylinder H, some water escaping from the plate-closing to the main cylinder through the common supply pipe to allow the main cylinder H to move relatively to the plate-closing cylinder K. After the pressure has been kept on the rivet a short time, the water from K and H is exhausted back into the tank, the pressure on the annular drawback piston E causing the return stroke on the other or full area end of the piston being opened to the exhaust.

Wherever possible the cylinders should be lined with gun-metal or phosphor bronze, the valves being also of the same metal throughout.

PART VII.—PUMPS.

CHAPTER XVI.

HAND AND POWER PUMPS.

IN examining briefly the ordinary forms of hand and power driven pressure pumps for transmitting water under pressure to presses, accumulators or other hydraulic machines, we pass over entirely the ordinary suction and bucket and plunger or force pumps used for the domestic supply and delivery water into tanks or reservoirs, and glance

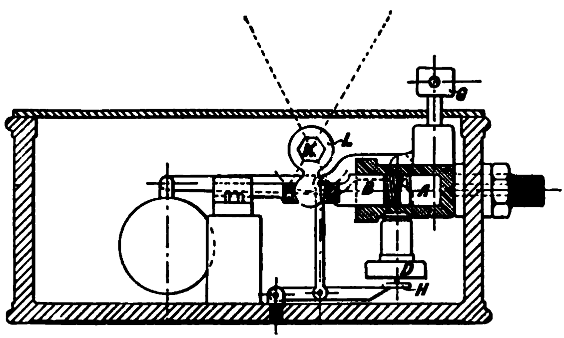


Fig. 132.

instead at the type of hand pressure pump as shown in sectional elevation in Fig. 132.

The pressure pump as shown is suitable for working up to pressures of 2 tons per square inch, and is particularly useful for boiler and other testing purposes, the pump A being fitted with a trip lever H for opening the suction valve upon the set pressure being obtained. The plunger B, Fig. 133, is reciprocated in the cylinder A of the pump casting, the water

entering from the cistern or tank to which the pump is secured through the suction valve C protected by a strainer D to fill the cylinder A. The back stroke of the plunger forces the water through the non-return valve F, closing at the same time the suction valve C, and delivering the water through the end branch E of the pump stem to the pipe attached thereto. To release the pressure the stop spindle G is turned, thereby opening the delivery port to an outlet port allowing the water to flow back into the cistern.

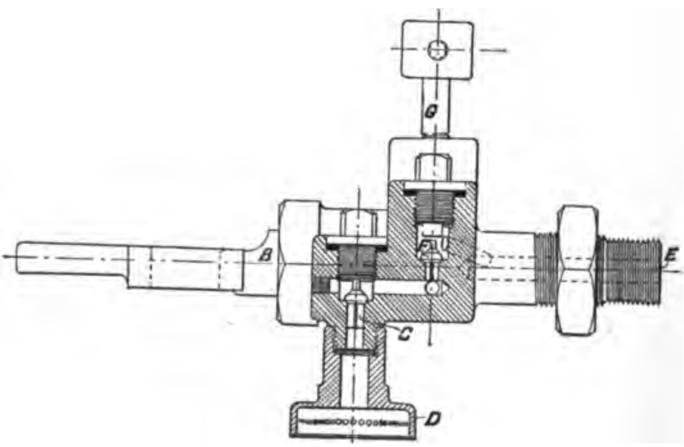


Fig. 133.

The plunger is reciprocated by a hand lever which is placed on the end of the spindle K, thus giving the desired movement to the tumbler or cam arm L, which works in an opening provided in the central portion of the plunger.

The passages for the water are drilled out of the solid metal of the casting, and the ends afterwards plugged by screwed and riveted plugs as shown in Fig. 134. The trip or release valve is described in connection with the vertical plunger pump shown in Figs. 136 and 137. The hand

pressure pump illustrated in Fig. 135 is provided with a vertical plunger A, and has the hand lever balanced and pivoted on to the standard or frame B, a trip or relief valve C is arranged upon the pump, and the stop or release valve D is placed horizontally. The passages and valves of the pump are similar to the valves shown in Fig. 133, the plunger also being of the same type, having its packing formed by a leather lace bound tightly round a groove.

Pumps driven by belting or gearing for hydraulic purposes

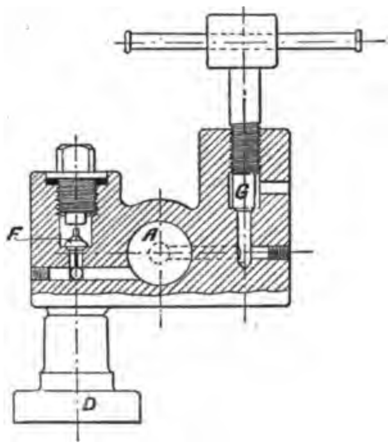


Fig. 134.

have much in common with the typical hand pump already examined, and Figs. 136 and 137 show in elevation and in detail a very useful type of belt-power pressure pump. The crank shaft is connected direct to the plungers, which are arranged in varying sizes upon the standard for the purpose of giving a quick run up of water at a low pressure for such a duty as a packing press where, as we have before pointed out, a varying pressure is always required during the travel

of the press to suit the density of the material which is being compressed.

Trip levers are connected with each pump, and they are so arranged that the pressure produced upon the water by the resistance of the material between the press platten

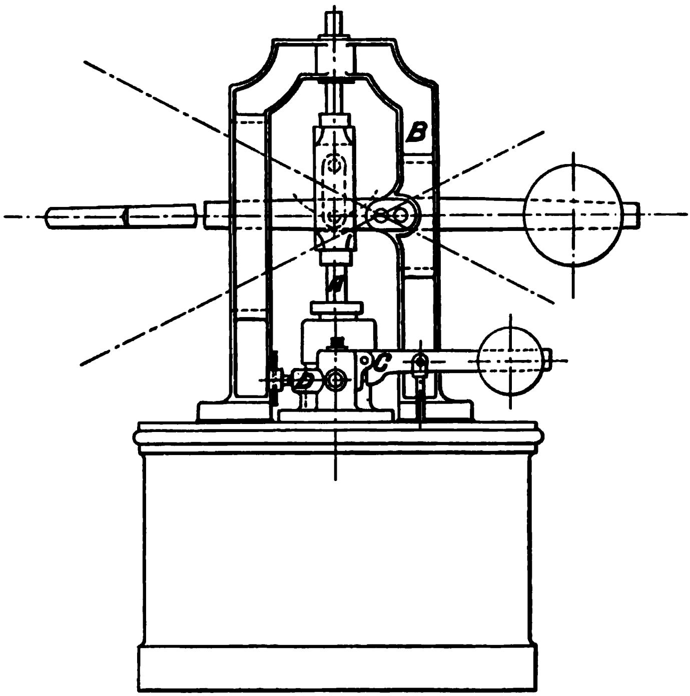


Fig. 135.

and head shall cause a small valve B to raise the loaded lever C, and with it the bottom foot lever D, which then raises the suction valve W off its seat, thus causing the power of the pump to be given to the two remaining plungers, which are of smaller area. When the pressure is further increased

owing to the material being more densely compressed, the second trip lever is caused to move by its valve being urged to overcome the corresponding weighted lever, and thus

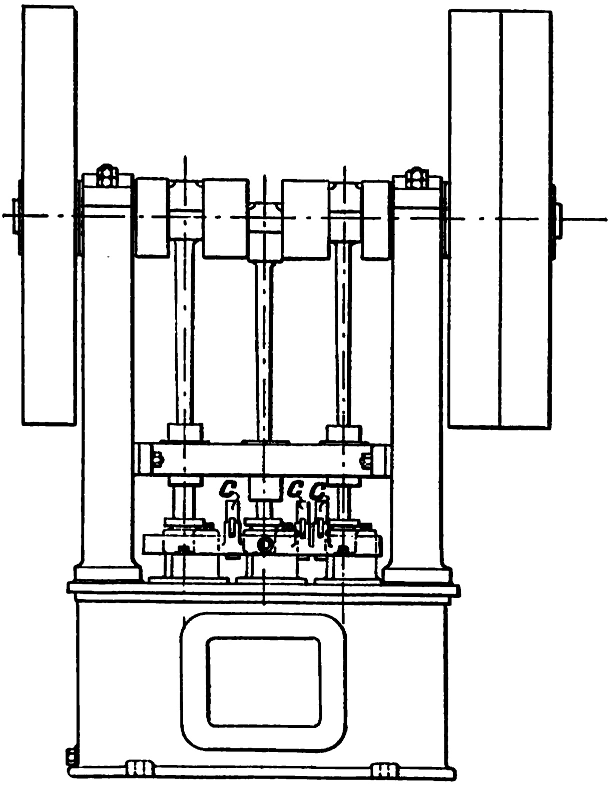


Fig. 136.

another plunger is thrown out of action, leaving the last plunger of a smaller diameter to give the final pressure to produce the maximum load against which it is set by its trip lever. By this arrangement of trip levers any desired pressure

can be produced upon the final plunger while leaving the early movements of the pump to deliver water at a very much lower pressure, thereby economising the power and water and making the pressing operation a quick one. It should be noted that the trip valve which acts against the loaded lever does not allow any water to escape, but simply moves upwards within its bored port, the leather packing on

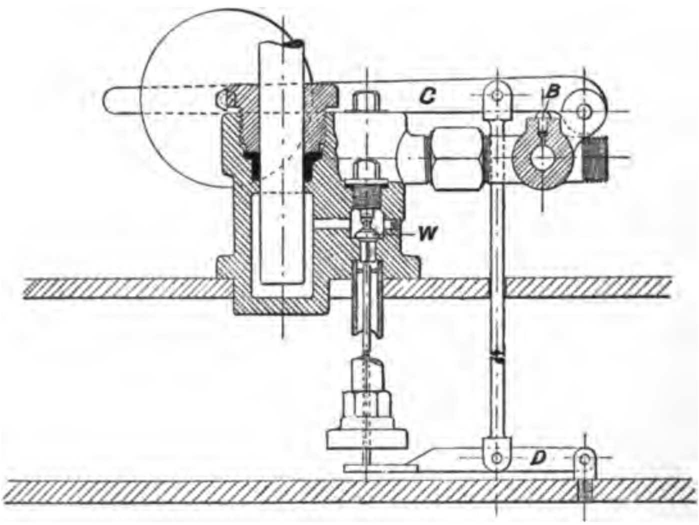


Fig. 137.

the end of the valve keeping the pressure tight within the pump passages.

The well-known bucket and plunger pump employed for ordinary water-raising purposes, where a continuous flow of water is required from the single up and down motion of one plunger, has its counterpart arrangement for hydraulic power purposes as shown in Fig. 138. In this pump, which is suitable alike for hand or power, the suction valve A is only

operated at each alternate stroke, and by proportioning the areas of the plunger half the quantity of water drawn in the suction valve A is delivered through the delivery valve B at each stroke. The non-return valve C, which acts as the check valve to the full end of the piston, is forced upon its seat during the in or suction stroke of the piston by the pressure water travelling from the annular or front end of the pump, the valve B being opened for delivery during this period. During the outward stroke of the piston the suction

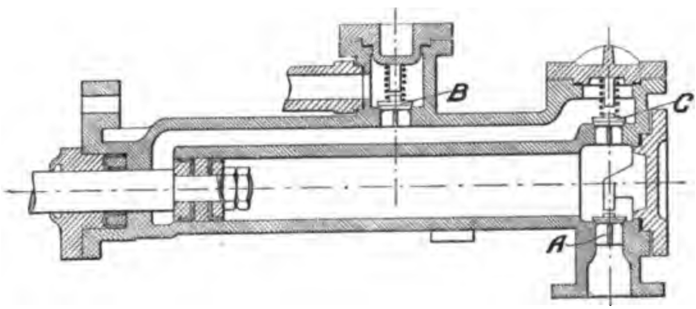


Fig. 138.

valve A is forced on to its seat, but the check valve C is raised, allowing the full bore of the pump barrel to be discharged through it, half of this quantity going to fill up the annular space in front of the piston, while the other half is delivered through the outlet valve B. This counterbalancing of fluid pressure within the pump barrel renders the arrangement particularly suitable for all classes of pumping machinery, as no unequal strains are set up during the working of the pump at any speed.

CHAPTER XVII.

STEAM PUMPS.

THE varieties of steam pumps for hydraulic pressure purposes are almost as numerous as the varieties of the ordinary steam engine, although possibly the pumps have more in common than have the engines produced by various makers.

Unquestionably the most satisfactory type for general

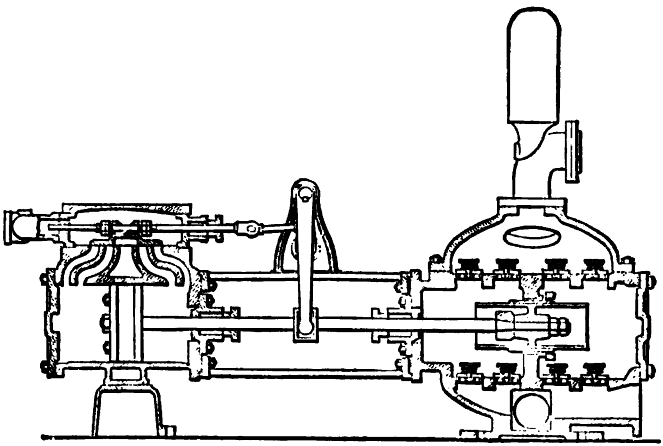


Fig. 139.

purposes of a small installation where steam is available is the duplex pump, first introduced and perfected by H. R. Worthington, of America. The Worthington pump, as illustrated in Fig. 139, has two steam cylinders side by side, the piston rods of each cylinder being continued to act as the pump rods of the two pumps at the opposite end, the pump

castings being connected to the cylinders by distance pieces, as shown. The valve of each steam cylinder is an ordinary slide valve, but the ports are duplicated at each end. No lap or lead is given to the valve, but a small space or slack is given between the nuts and the jaw of the valve. This lost motion permits the valve rod to travel slightly before moving the valve, thus allowing a slight pause in the motion of the piston at the end of each stroke, thereby giving the water valves time to seat smoothly and without violence. The valve of one cylinder is controlled by the piston rod of the other, the motion being transmitted through the vibrating arm pivoted on the distance piece. The moving parts being

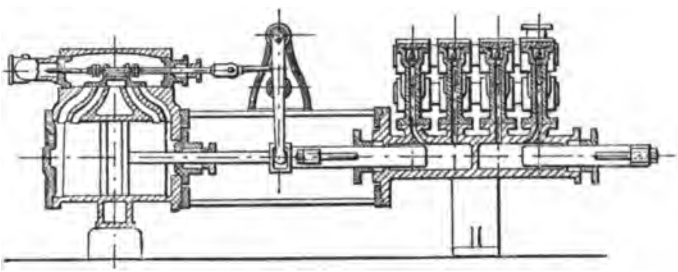


Fig. 140.

always in contact, the blow which arises with tappet controlled valves is avoided. When the piston in its motion covers the first port, which is the exhaust, the steam remaining in the cylinder is cushioned in front of the piston, thus causing a gradual arrest of its movement. One or other of the slide valves being always open, there is no dead point, and the pump is therefore capable of being stopped and started at any time. This property of constant readiness for full duty enables the Worthington or duplex pump to be employed for working direct on to hydraulic lift cylinders or on to an accumulator, the pump following up the motion of the lift on the rise and fall of the accumulator automatically

when the pressure from the pump delivery main is drawn upon. In connection with pumps it is desirable to employ an air chamber on the suction main as well as on the delivery main, in order to make the flow of water continuous and to ensure that the cylinder shall be filled at each stroke. When an air vessel is not possible on the suction side, it is an advantage to give the water entering the valve a little head by causing a T branch connection with the suction pipe and the pump barrel to be made, the water in the T thus standing above the pump barrel. The flow into the suction pipe should not exceed 150 to 200 feet per minute. The speed of the plunger may be from 65 to 150 feet per minute.

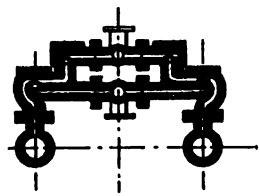


Fig. 141.

The pressure pump shown in section in Fig. 140 is a Worthington packed plunger or double-ram pressure pump.

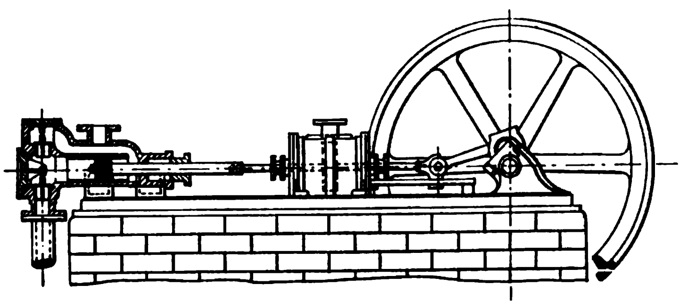


Fig. 142.

The barrel is divided, so that each end is an independent single-acting plunger drawing water at the one end, while the opposite plunger is forcing it out at the other end of the divided barrel. A number of independent pressure valves

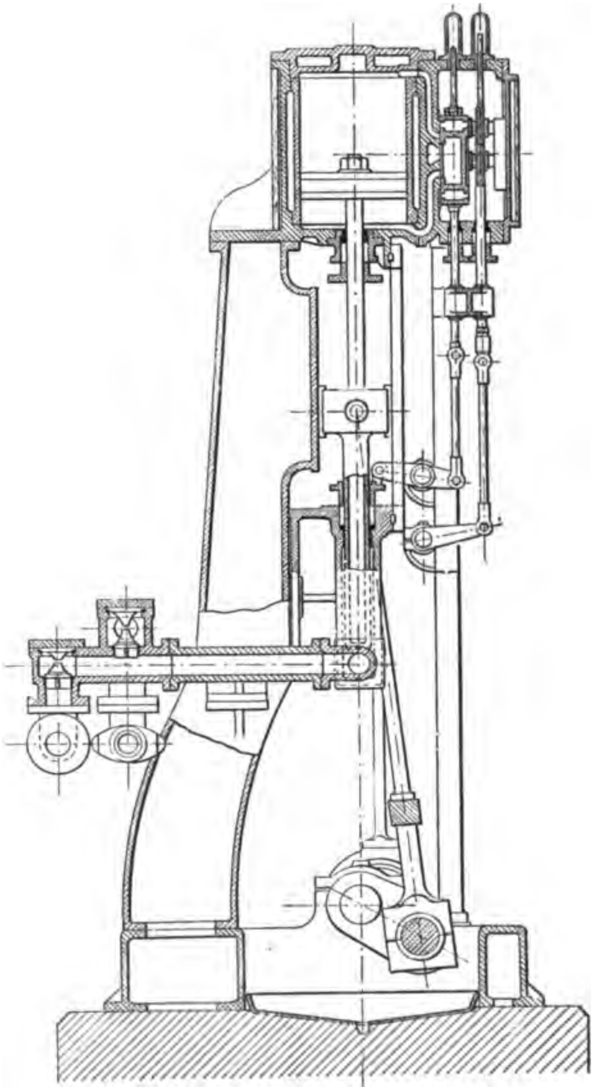


Fig. 143.

are employed, easily accessible, and are contained in small chambers for resisting heavy pressures. These pumps work up to 8,000 lbs. to the square inch. The plungers are connected by means of yokes and outside rods, so that they move together as one plunger and become double acting by the division of the barrel. Fig. 141 shows a sectional view of the pump barrels and their valves, a common suction and delivery branch being alone required for the two independent double-acting pump barrels. These pumps work best when the plunger speed does not exceed 50 feet per minute.

A fly-wheel double-acting pressure pump, having a hori-

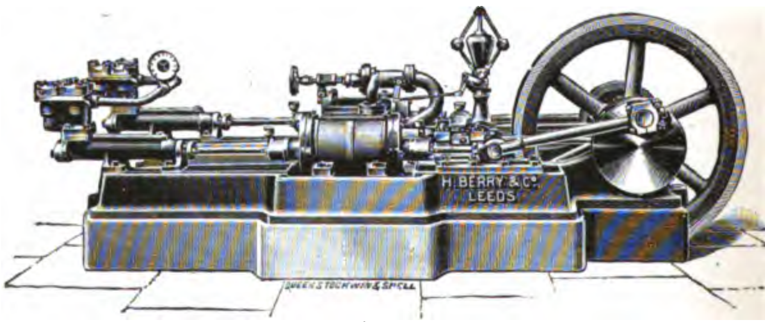


Fig. 144.

zontal steam cylinder, as shown in Fig. 142, is often employed for small hydraulic installations. The valves are arranged at the extreme end of the pump, and being immediately above each other, admit of easy examination and renewal.

A vertical cylinder engine with expansion valve having direct coupled pumps is shown at Fig. 143. In this example the water is drawn in and forced out at right angles to the line of axis of the pump. The valves are very accessible, and the pump plungers are easily packed. This type of engine is in use at the pumping station of the Hydraulic Power Company, of London.

The pumps illustrated in Fig. 144 were made by Messrs Berry for the London County Council, and have two steam cylinders with direct-acting pumps, $2\frac{1}{4}$ inches diameter by 12 inches stroke, the pump plunger rods being connected through the back ends of the cylinders to 9 inches diameter steam pistons. The pumps supply an accumulator, and work at 750 lbs. per square inch.

PART VIII.—HYDRAULIC MOTORS.

CHAPTER XVIII.

TURBINES.

Before proceeding to the detailed examination of the various types of turbines, we will examine the action of a stream of water on a curved vane. If a stream of water having a certain velocity c_1 meets a stationary curved vane, the path of the stream will be altered, following the curve of the vane and leaving in the direction which the vane would take if continued. Neglecting losses from friction, the velocity c_2 of the stream will be the same on leaving the vane as on entering, the only change being one of direction. If now a velocity w_1 be given to the vane, an inspection of the diagram (Fig. 145) will show that the water may never touch the vane at all; for when the stream has reached c_3 the vane will have travelled to w_2. To obviate this, either the orifice of the stream must be given a motion similar in direction and magnitude to w_1, or the direction and velocity of the stream must be altered to c, the resultant of c_1 and w_1. The motion of the stream relative to the moving vane again coincides with c_1. The motion of the stream on leaving the vane will again coincide with c_2 relatively to the vane, but as the vane and stream each have the velocity w_1, the absolute or real velocity of the stream on leaving the vane will be the resultant of c_2 and w_1, or u. On entering the vane the stream had an absolute velocity of c, and a corresponding store of energy—

$$\left(\frac{Wc^2}{2g}\right)$$

On leaving the vane the absolute velocity of the water is u and the corresponding energy—

$$\left(\frac{Wu^2}{2g}\right)$$

Now if u is less than c the energy remaining in the water on leaving the vane must be less than the original energy contained in the stream, so that neglecting the losses by friction the difference of energy has been imparted to the vane, and is capable of being applied to perform useful work.

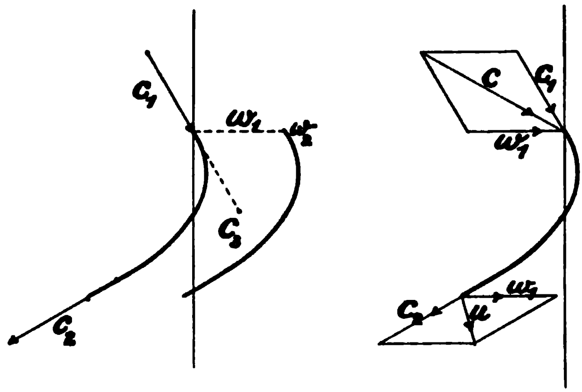


Fig. 145.

The velocity of entry c is generally fixed by circumstances, and the designer has to convert as large a percentage of the energy contained in the stream at disposal into useful work. This is obtained by keeping the velocity u of discharge as low as possible, and thereby increasing the difference between the energy of the entering stream and that of the leaving stream. The velocity u cannot in practice be made O, as the water would not then flow from the vane at all.

It will be noticed that no mention has been made of the exact curve a turbine vane should take, and it may be here

stated that there is no particular curve to be followed, the only conditions being that the curve of the vane shall flow gradually from the angle of entry to the angle of exit.

There are two distinct classes of turbines, namely, *Impulse* and *Reaction*. Each of these classes contains several types, having the flow of the water arranged in different directions. These types may be enumerated as below :—

IMPULSE.	REACTION.
No Suction Tube.	*With or without Suction Tube.*
Radial outward flow.	Radial outward flow.
,, inward ,,	,, inward ,,
Axial flow.	Axial flow.
Pelton wheel.	

In an impulse turbine the action of the stream follows very closely the explanation already given, and our subsequent remarks will relate more to precautions to be observed in designing. The water is directed into the vanes of the wheel in the required direction by fixed guide vanes, so arranged in size that the wheel is never allowed to become filled with water or drowned. The outlet is also above water, so that the stream in passing through the turbine is at all times under atmospheric pressure.

Fig. 146 shows a sectional elevation of a Girard impulse turbine, and Fig. 147 shows an end elevation partly in section of the same wheel. The water enters through the pipe A, and passing through the regulator valve B, is directed by the guide vanes C into the wheel vanes or buckets D at the correct angle for preventing shock from impact. After passing through the wheel buckets the water falls away at as low a velocity as circumstances will permit through the opening or tail-race E. The supply of water is regulated by the hand-wheel attached to the screw F which operates the lever G, and so causes motion of the slide valve B, which admits the water to the required number of guide passages. This

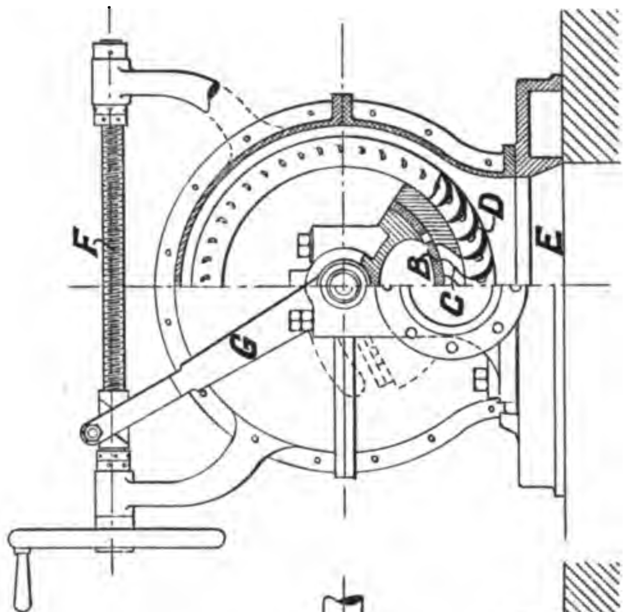


Fig. 147.

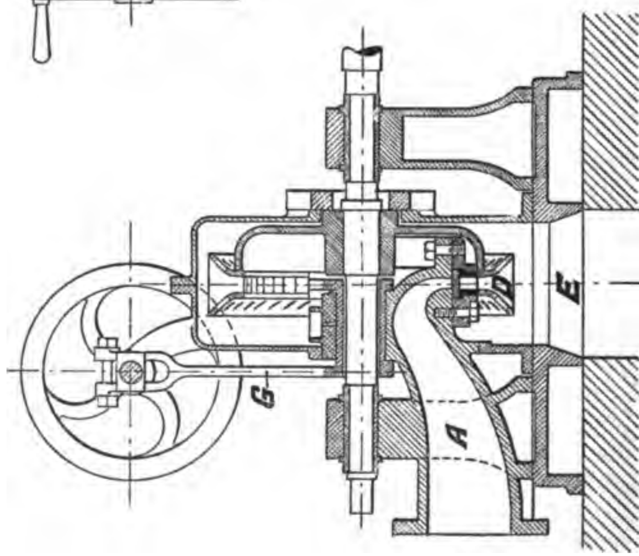


Fig. 146.

method of governing is well adapted to impulse turbines, and has no appreciable effect on the efficiency.

Fig. 148 illustrates the general arrangement of a Pelton wheel, which is a type of axial flow impulse turbine. The water leaves the jet A at a velocity dependent upon the head of water available, and meets the cups or buckets on

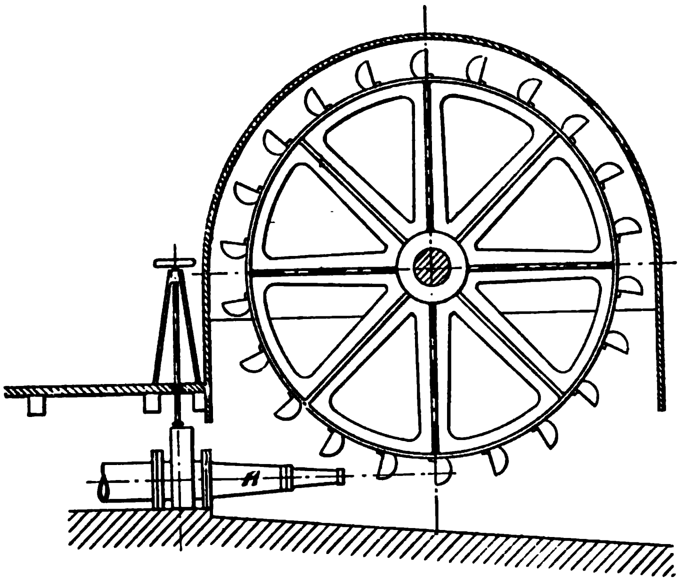


Fig. 148.

the wheel rim with as little shock as possible. The buckets are in the form of two hemispheres, joined together at the centre by a straight thin rib. The water meets the rib, and is divided into two streams, one going each way and acting on the curved surfaces of the buckets as the stream of water does in any other form of impulse turbine. The speed of the wheel should be such that the water on discharge from the buckets is almost stationary.

Fig. 149 shows an axial flow reaction turbine, which, though much like an impulse turbine in general appearance, is so proportioned and erected that the vanes are always full of water or drowned, and the water is discharged under the water level of the tail-race. The action of the water on

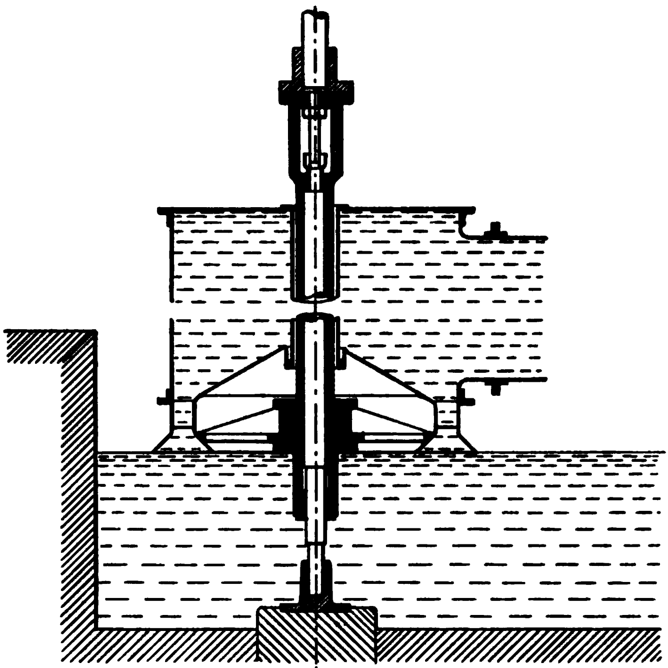


Fig. 149.

the vanes is similar to that given in the general explanation, but the velocity of the water through the wheel is not necessarily uniform, but depends on the sizes of the openings for outlet from the fixed guide vanes, also the outlet from the wheel vanes. Where the openings are narrow,

the velocity is correspondingly great, and where wide, correspondingly small, as in a pipe of varying diameter.

Reaction turbines are frequently fitted with suction tubes which permit of the wheel being placed at a height above the tail-race level dependent on conditions to be afterwards explained. The suction tube may alter the velocity of flow through the wheel according to its area of outlet and the pressure energy remaining in the water at the time of outflow.

Fig. 150 shows the usual type of thrust-bearing used in turbines having a vertical shaft.

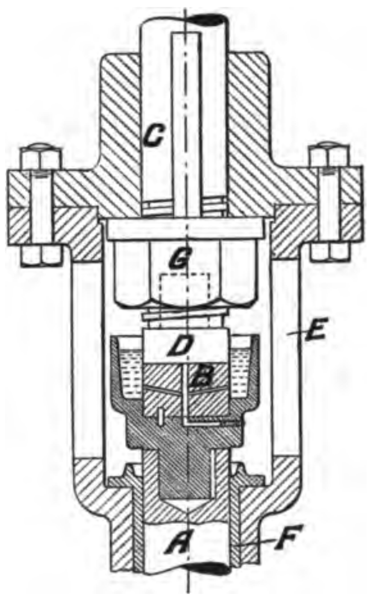


Fig. 150.

The arrangement will be better understood after an examination of Fig. 149. The vertical shaft A rests on a massive foundation, and carries at its upper end a fixed oil cup which contains the fixed steel block B. The mainshaft C carries a gun-metal block D which rests on the block B. The mainshaft C passes through a plummer block not shown in the figures, which provides lateral stability. The turbine wheel is supported by a hollow cast-iron shaft suspended from the main shaft C by the lantern E, which carries a brass bush F for steadying the upper end of the vertical shaft A. The turbine wheel is supported laterally by a brass bush carried by the lower end of the hollow cast-iron shaft, and fitting the vertical shaft A.

The nut G allows the turbine wheel to be adjusted vertically to compensate for the wear of the thrust block D.

There is an immense variety of turbines, but the more important types are—(1) The Fourneyron turbine, in which the water flows from within the wheel outwards, and at right angles to the axis; (2) the centre vent turbine, in which the water flows from the outside of the wheel towards its centre, also at right angles to its axis; (3) the Jonval or parallel flow turbine, in which the water flows through the wheel parallel to the axis; and (4) partial turbines, which may be of either of the other types, but in which the water flows into the wheel only round a portion of the circumference.

In all turbines the water is conducted by a set of fixed guide curves or plates into the revolving wheel, where it meets with buckets or curved partitions against which it impinges, causing the wheel to revolve.

CHAPTER XIX.

IMPULSE TURBINES.

In designing a turbine to utilise the energy of a supply of water under a head or pressure, there must be known the quantity of water flowing, and the head or pressure available. The fullest particulars as to variation of supply, highest flood levels, minimum supply during summer months, should also be ascertained if the proposed turbine is to meet the requirements to the best advantage. Where the fall is great and the quantity of water small, the choice must be in favour of an impulse wheel with partial admission, as a reaction turbine would require to be so small and to work with such a high number of revolutions that the design would become unsuitable if not impossible. If the head or fall is only a few feet, and the water supply fairly regular, as is the case where a reservoir or pound is used, a reaction turbine is very suitable, as it is not affected by change of level in the tail-race caused by flood, provided there is a corresponding rise in the top level; whereas an impulse turbine would require to be placed at a sufficient height above the level of the tail-race as to ensure that the flood shall never reach the wheel.

The chief objection to the reaction type as frequently constructed is the inability to economically supply varying power; so long as the power is the same that the turbine was designed to supply, a very good performance may be expected, but if a greater or less power is required the efficiency falls off rapidly. It will be seen that many reaction wheels are unsuited to a situation where the water supply falls short in dry weather, as if the wheel is designed to give

good results for high powers, the power given out with a limited supply will fall so much as to be practically useless. On the other hand, if the wheel is designed to be economical at low powers, it will never give out large powers, although there may be a large water consumption. Reaction turbines have been used in conjunction with impulse turbines, in which case the reaction wheel is set to work at its most economical power, whilst any alteration in power is obtained by regulating the supply to the impulse wheel.

Before commencing the design of an impulse turbine, the actual velocity of the water at the guide passages must be ascertained. If the water enters the guides from a long

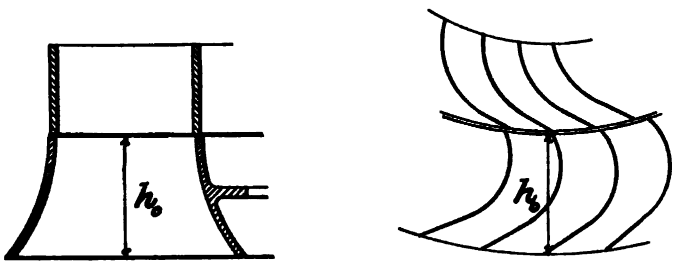


Fig. 151.

pipe or open channel and vertical pipe, having already discussed the formulæ in a previous chapter, we can calculate the actual effective head h, after allowing for frictional and other losses. This head should be calculated from the outlet level of the guide passages, allowance being made for the height h_0 above the tail level to allow for the buckets of the wheel, as shown in Fig. 151.

The velocity c of flow from the guide passages will then be—

$$c = .95\sqrt{2gh} \qquad (1)$$

.95 being the value of a coefficient taken from actual observation.

The next step is to find the total outlet area of the guide passages necessary to pass the maximum quantity of water. If the area were only made large enough to pass the quantity of water flowing with the velocity c, it would be found that the full quantity would not flow, as there is a certain amount of obstruction from the vanes passing across the guide passages. A smaller velocity is assumed in calculating the area of the openings having a value of $.89c$, so that the formula becomes—

$$A = \frac{Q}{.89c} \quad - \quad - \quad - \quad - \quad - \quad - \quad - \quad - \qquad (2)$$

in which Q represents the quantity of water in cubic feet per second, and A the required area in square feet. We have now two more dimensions to settle, namely, the width of the buckets and the radius of the wheel; either of these can be adjusted to requirements by an alteration of the other. Before proceeding further a trial radius should be decided upon, also the angles a and a_2 (see Fig. 152).

The wheel velocity is fixed between narrow limits by the velocity of entry of the water if the turbine is to be a really efficient machine; as is also the angle a of entry.

We will try to explain the reason for this by the aid of the diagrams (Fig. 152).

We have already observed in our preliminary remarks that the less the value of u, the velocity of exit, the greater the efficiency of the turbine, while the direction of inlet does not of itself affect the efficiency, except that no turbine has yet been designed in which the velocity of u can be regulated without adjusting the angle of inlet a.

From the point o draw c, representing to scale the absolute velocity and direction of the stream passing through the guide passages of a turbine. Draw c_1, as shown, and complete the parallelogram by drawing w_1, the wheel velocity. For the present argument we will assume that the velocity c_2 of exit is the same as c_1, and that w_2 is equal to w_1.

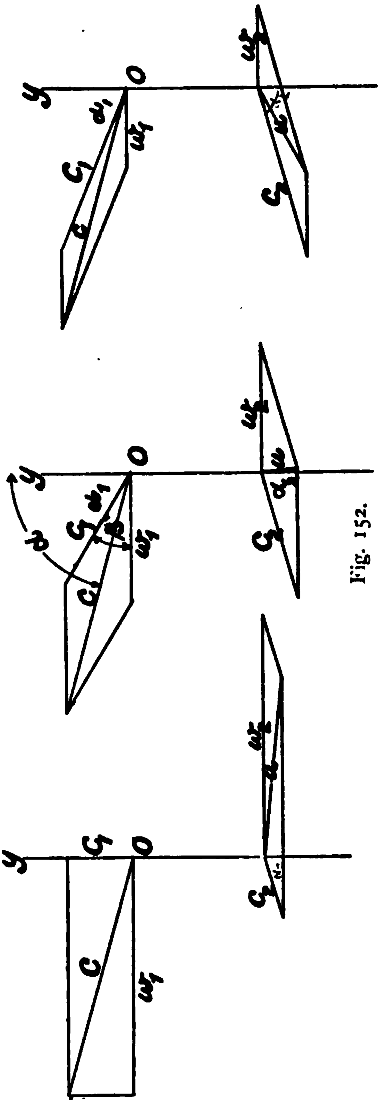


Fig. 152.

The direction of w_2 must of necessity be parallel to w_1, while the direction of c_2 may be altered at will. Select a direction for c_2, making any angle α_2 with the ordinate Oy; complete the parallelogram, and obtain the corresponding value of u. In all the diagrams the angle α_2 has the same value. In the first diagram, by selecting a vertical direction for c_1, and consequent value of $\alpha_1 = 0^\circ$, the value of c_1 is small, whilst w_1 is large, giving u a forward direction and high velocity.

In the second diagram c_1 and w_1 have been made equal to each other, and the angle β $(= 90^\circ - \alpha_1)$ is consequently bisected by the line c. c_2 and w_2 being the same in value as c_1 and w_1, are equal to each other, so that u will have a slightly forward direction and small value.

In the third diagram c_1 has a large value, and w_1 a small value, so

that on drawing out the parallelogram c_2 w_2 the velocity u is found to have a large value in a backward direction. Now, as we have previously shown that u should be as small as possible, it is evident, without further demonstration, that c_1 should be slightly greater than w_1, and consequently c_2 greater than w_2. To what extent this rule may be followed in practice, and the modifications necessary in the various designs of inward, outward, or radial flow turbines, will be further explained.

In an axial turbine the value of w_2 being the same as w_1, it would appear at first sight that the conditions above stated apply without correction; but this is not so, as, owing to the height h_o (Fig. 151), the stream of water will increase in velocity in passing through the vanes, the additional velocity being represented by $\sqrt{2gh_o}$; but as there is friction between the vanes and the stream, the velocity of the water will be reduced below the theoretical amount, so that the complete formula becomes—

$$c_2^2 = (c_1^2 + 2gh_o)\frac{1}{1+f} \quad - \quad - \quad - \quad - \quad (3)$$

The value of f is variable between .05 and .1. The value of h_o cannot yet be fixed, so that in calculating c_2 an assumption must be made, 6 inches to 1 foot being a suitable dimension. It is scarcely necessary to remark that with high falls, and consequently high velocities, h_o may be neglected in the preliminary calculations, as its effect becomes scarcely noticeable; whereas with a low fall the height h_o forms a considerable portion of the total head.

In an inward flow radial turbine w_2 is less than w_1 by an amount dependent upon the ratio of the depth of the vane to the radius, and as c_2 should be slightly greater than w_2, the value of c_1 (greater than c_2) may be temporarily fixed approximately equal to w_1. Fig. 153 will make this clear.

Fig. 154 shows the diagram for an outward flow radial turbine, in which w_2 becomes greater than w_1 by an amount

dependent upon the ratio of the depth of the vane to the radius. c_2 must be slightly greater than w_2, and consequently considerably greater than w_1.

Having arrived at suitable values of c_1, w_1, and a and a_1, we may calculate the width of the vanes necessary to pass

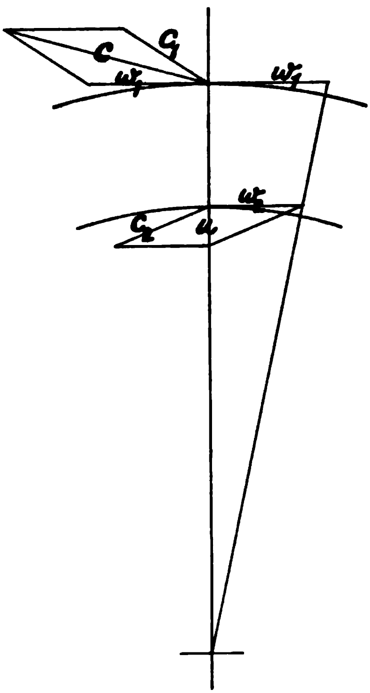


Fig. 153.

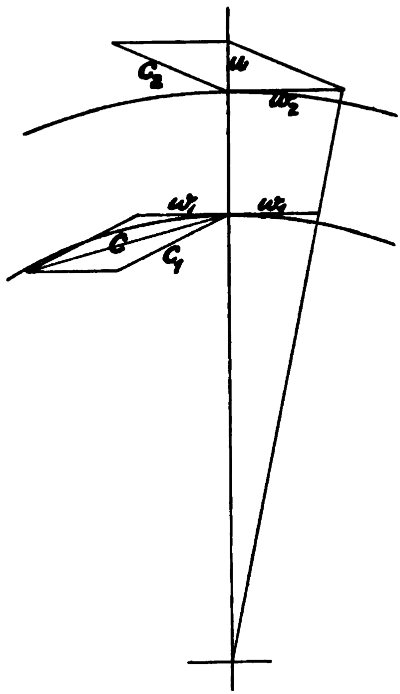


Fig. 154.

the quantity of water flowing. The values of A, the area of outlet, and r, the radius, being known, we have the following formula—

$$e_1 \text{ (width of vane)} = \frac{A}{2\pi r. \cos. a - z_1 t_1} \quad - \quad - \qquad (4)$$

in which z_1 is the number of guide vanes and t_1 their thick-

ness. The width e_2 of outlet from the vanes may be calculated in the same way by the formula—

$$e_2 = \frac{A_2}{2\pi r_2 \cos. \alpha_2 - z_2 t_2} \quad - \quad - \quad - \quad - \quad (5)$$

in which A_2 represents the area necessary to pass the quantity Q of water flowing with the velocity C_2; r_2 represents the radius at outlet, t_2 the thickness, and z_2 the number of vanes. z_2 should always be less than z_1, so that the vanes shall not be choked with water, and so interfere with free deviation. For the same reason the width of the wheel vanes should be made larger than the value given by the above equation.

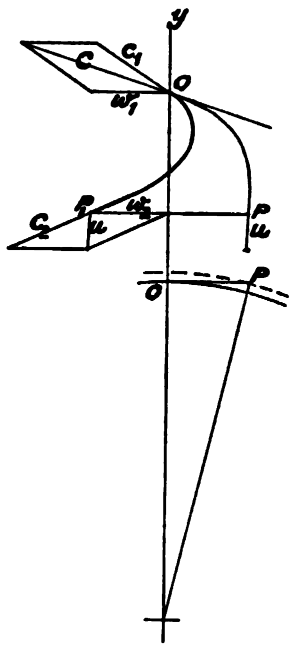


Fig. 155.

If the radius chosen gives unsuitable values for e_1 and e_2 a new radius must be selected, and the calculations repeated. If, however, the value of e_1 comes out too small, partial admission should be resorted to.

The values chosen for the angles α and α_2, if too large, will give trouble, and must be reduced, and a new trial made.

The above remarks apply equally to all classes of impulse turbines. There are, however, two more points to be considered in connection with axial flow turbines—namely, the centrifugal effect of the water due to the fact that the stream enters the vanes in a tangential direction, whilst the vanes move in a circular path; also owing to the fact that all

parts of the vane are not at the same radial distance, the quantities w_1 and w_2 have variable values.

The centrifugal effect may be easily counteracted, as will be seen with reference to the diagram (Fig. 155). The values of the angles a_1 and a_2 having been fixed, and the design of the turbine completed in every way, the absolute path of the stream of water through the turbine may be easily found by measuring the length of the turbine vane in terms of c_1; now mark off P_1 P the same multiple of w_2, and the point P indicates where water entering at the point O would leave the vanes. If, for example, the length of vane equals $2 \times c_1$, then a distance equal to $2 \times w_2$ must be marked off. The circumference of the vanes must now be drawn to

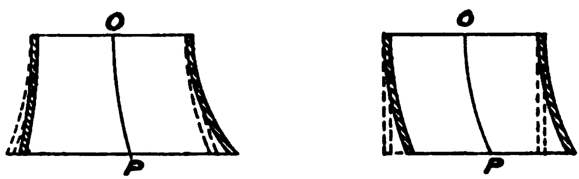


Fig. 156.

scale, and the distance O P marked off tangentially from O will indicate the correct radius of the wheel where the water leaves. If the value of c_2 differs greatly from c_1 the mean value should be taken in making the above calculation. The vanes may now be corrected in shape, as shown by full lines in Fig. 156.

With regard to the effect of the varying radius, and consequent variation of w_1 and w_2, we have only to turn back to the diagrams (Fig. 152) to see the result. The design should be prepared with reference to the mean radius, when the outer radius will give a diagram similar to the first, and the inner radius a diagram similar to the third in Fig. 152. To get the best effect the curve of the vane must be gradually changed to suit the varying values of c_1 and w_1. An

inspection of Fig. 152 shows that the direction of w will be forward at the outer radius, and backward at the inner; consequently in designing an axial wheel the radius should be as large as possible, and the width e of the vanes as narrow as possible.

We will now consider the Pelton wheel, which is a special form of axial flow turbine, having the angle $\alpha = 90°$, and consequently $c_1 + w_1 = c$; and, as we have already explained, c_1 should also equal w_1. We find that in a Pelton wheel the velocity w_1 should be half the actual velocity C of the water issuing from the jet. The angle α_2 cannot be made equal to 90°, as the water would strike the next bucket. There is a certain amount of impact where the jet of water meets the thin edge of the bucket, as it is impossible to make a sharp edge in practice. The chief advantages claimed for the Pelton wheel are its simplicity of construction, which renders it particularly suitable for transport in new countries, and its high efficiency.

CHAPTER XX.

REACTION TURBINES.

IN order that the theory of reaction turbines may be made clear, we will start our investigation by reconsidering the design of impulse turbine examined in our last chapter. Referring back to Fig. 151, the vanes might easily be so designed by properly proportioning the width e_1 and e_2, that the area A_2 is equal to the area A_1 and consequently greater than the area A, measured in a direction at right angles to the direction of flow as shown in Fig. 157. If this is done, the water will still have the velocity c_2 on leaving the wheel, but the buckets will be filled with water at the inlet and outlet. The correct velocities c, c_1, c_2 are shown in Fig. 157, and are the same as for an impulse wheel. If the outlet be now placed under water as in Fig. 149, the wheel will become filled with water at all parts, and, neglecting the slight variation in the frictional losses due to the altered conditions, will have the action of a free deviation impulse turbine.

The design of the turbine may now be altered so that the area A_2 has a larger or smaller value than that given by the conditions of Fig. 157, and as the wheel is at all parts full of water, the velocity of flow at any point is governed by the formula $Q = Av$. By altering the area A_2 we not only alter the velocity c_2, but also the velocities c and c_1, and, whereas the velocity c for impulse turbines has one particular value for any given head of water, the velocity c for reaction wheels may have a comparatively large range of values for any given head.

Fig. 158 shows what takes place if the value of e_2 is the

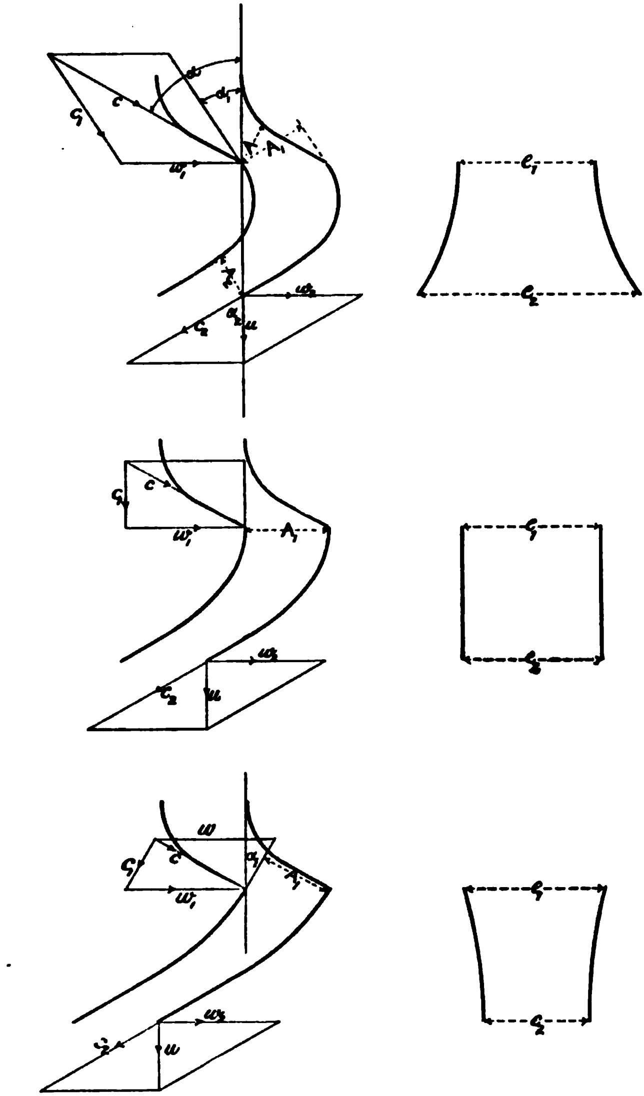
same as c_1, or if the vanes are of the same width throughout. In the Figs. 157, 158, 159, we have taken the same values for a and a_2 for the sake of comparison, while a has also been taken equal to a_2. In Fig. 158, A_2 will consequently equal A, and c_2 will equal c. Taking the value for w_2, which makes u vertical, we see that c_1 enters the vanes in a vertical direction. This diagram is typical of the design of the Jonval turbine as conducted on the European continent.

In Fig. 159 the wheel vanes have been contracted, causing diminution of the area A_2 in relation to A, and consequent increase of the velocity c_2 above c. Applying the correct value for w_2, c_1 is given a backward direction. Thus we see that for any values of a and a_2 by altering the ratio of the areas A and A_2 we can produce different values of c for the same head of water. In the diagrams the same length of line has been taken to represent the value of c_2, but it must not be supposed on this account that c_2 has the same arithmetical value in each case. As we have not yet investigated the formulæ for calculating the true value of c under any conditions, some value had to be assumed in order that the diagrams could be drawn out, so that, while in each diagram the values of c, c_1, c_2, w_1, w_2, and u are proportional to the lengths there given, the diagrams must not be compared by measurement.

The correct value of c for any conditions must next be investigated. As the turbine is filled with water, and the flow at any point is governed by the formula $Q = Av$, the energy contained in the water at any point is evidently represented by the hydrodynamic equation already investigated in Chapter I., or, if h represents the useful head of water, the energy of 1 lb. of water is—

$$h = h_1 + \frac{v_1^2}{2g} = h_2 + \frac{v_2^2}{2g} = \text{etc.}$$

If h_1 represents the pressure energy of the water on leaving the guide passages, then the total energy of the water on



Figs. 157, 158, and 159.

leaving the guide passages is $h_1 + \frac{c^2}{2g}$. Now this energy, neglecting losses, must balance the energy h_m of the total head of the water, measured from its surface to the level of outflow from the buckets, as shown in the diagram Fig. 160, therefore—

$$h_m = h_1 + \frac{c^2}{2g} \quad - \quad - \quad - \quad (1)$$

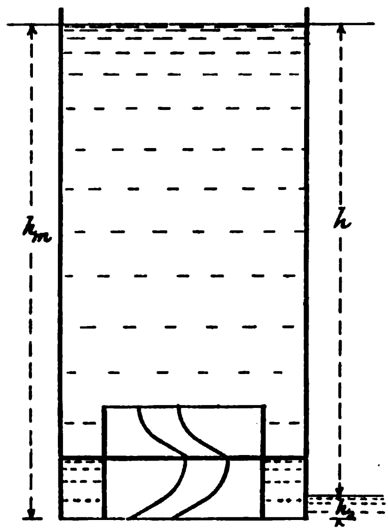


Fig. 160.

As, however, the water level in the tail-race is liable to vary and rise a height h_2 above the outflow level, the useful head h is evidently represented by $h_m - h_2$, so that from equation (1) we get—

$$h = h_m - h_2 = h_1 - h_2 + \frac{c^2}{2g} \quad - \quad - \quad (2)$$

We must now consider what is taking place in the turbine

buckets due to the change of velocity from c_1 to c_2. From the hydrodynamic equation—

$$h_1 + \frac{c_1^2}{2g} = h_2 + \frac{c_2^2}{2g} \quad - \quad - \quad - \quad (3)$$

therefore

$$h_1 - h_2 = \frac{c_2^2}{2g} - \frac{c_1^2}{2g} \quad - \quad - \quad - \quad (4)$$

By substituting this value for $h_1 - h_2$ in equation (2) we get—

$$h = \frac{1}{2g}(c^2 + c_2^2 - c_1^2) \quad - \quad - \quad - \quad (5)$$

The values of c_2 and c_1 may now be expressed in terms of c, since—

$$c_2 : c :: A : A_2$$

$$\therefore \quad c_2 = \frac{A}{A_2}c \quad - \quad - \quad - \quad - \quad (5a)$$

similarly

$$c_1 = \frac{A}{A_1}c \quad - \quad - \quad - \quad - \quad (5b)$$

Substituting these values in equation (5)—

$$h = \frac{1}{2g}\left\{c^2 + \left(\frac{A}{A_2}\right)^2 c^2 - \left(\frac{A}{A_1}\right)^2 c^2\right\}$$

$$= \frac{1}{2g}.c^2\left\{1 + \left(\frac{A}{A_2}\right)^2 - \left(\frac{A}{A_1}\right)^2\right\} \quad - \quad - \quad (6)$$

$$\therefore \quad c^2 = 2gh.\frac{1}{1 + \left(\frac{A}{A_2}\right)^2 - \left(\frac{A}{A_1}\right)^2}$$

$$\therefore \quad c = \sqrt{2gh}.\sqrt{\frac{1}{1 + \left(\frac{A}{A_2}\right)^2 - \left(\frac{A}{A_1}\right)^2}} \quad - \quad (7)$$

$$= K\sqrt{2gh} \quad - \quad - \quad - \quad - \quad - \quad - \quad (8)$$

The solution of equation (7) will give the value of c for the corresponding values of A, A_1, and A_2. The equation in this form is not suitable for direct use, as, though we may

assume values for A and A_2, the value of A_1 is entirely dependent of the values of A and A_2 and the angles α and α_2. So that having selected the values of A, A_2, α and α_2 is necessary to draw out a diagram similar to Figs. 157, 158, or 159, and so obtain the corresponding values of A_1 and α_1.

In drawing out the diagram c_2 should first be drawn in the correct direction making the angle α_2 with the vertical, and having a length not less than 1 inch, preferably 2 inches. w_2 must now be drawn so as to give u a vertical direction. The length of c may now be calculated from equation (5*a*) and drawn in a direction making the selected angle α with the vertical. On drawing w_1 equal to w_2 for an axial flow turbine and completing the parallelogram the value and direction of c_1 are obtained. A_1 may now be calculated from equation (5*b*).

Equation (7) may now be solved, and the value of c obtained, whence the other values, c_1, c_2, w_1, w_2, may be obtained either graphically or by calculation. This completes the calculation necessary in the case of an axial flow turbine working under the conditions assumed in the Figs. 157, 158, 159. If the value of u resulting is considered too high, then the process must be repeated with an altered ratio of A : A_2, and if necessary altered values for α and α_2.

If the calculation has to be made for an outward or inward flow radial wheel the only altered condition is in the value of w_1, which will not equal w_2, but will have a greater or less value. If r_1 and r_2 be the radii corresponding to w_1 and w_2 respectively, then

$$w_1 : w_2 :: r_1 : r_2$$

$$\therefore w_1 = w_2 \cdot \frac{r_1}{r_2} \qquad - \quad - \quad - \quad (9)$$

This new value for w_1 must be used in drawing out the diagram, and consequently c and c_1 will have an altered ratio to c_2. This operation may be performed graphically, as in Figs. 153, 154, Chapter XIX.

As it is not always convenient to resort to graphic methods, we may evolve an equation from equation (6) in which A_1 is expressed in terms of A, A_2, sin a and sin a_2.

Referring to any of the diagrams, Figs. 157, 158, 159, w'_2 may be expressed—

$$w'_2 = c_2 \sin a_2 \quad - \quad - \quad - \quad (10)$$

therefore by (9)—

$$w_1 = \frac{r_1}{r_2} w'_2 = \frac{r_1}{r_2} c_2 \sin a_2 \quad - \quad (11)$$

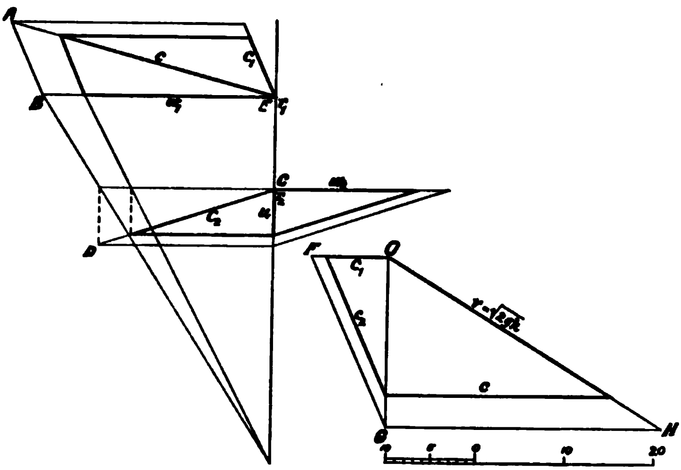


Fig. 161.

The whole operation of fixing the values of c, c_1, c_2, w_1, w'_2 may be performed by the following very simple graphic method :—

Having selected a and a_2, and the ratio $r_1 : r_2$ in the case of inward or outward flow turbines, draw out the diagram to any scale as shown by the light lines in Fig. 161. Set up a vertical OG and draw OF at right angles equal to AB in the diagram, and with the length CD mark off FG completing

the triangle OFG. From G set off GH at right angles to OG and equal to AE.

Join OH. Then $OF = c_1$, $FG = c_2$, and $GH = c$ to any scale.

$$\text{And } FG^2 - OF^2 = OG^2,$$
$$\text{or } c_2^2 - c_1^2 = OG_2.$$
$$\text{Again } GH^2 + OG^2 = OH^2,$$
$$\text{or } c^2 + c^2_2 - c_1^2 = OH^2.$$

From equation (5)—

$$c^2 + c_2^2 - c_1^2 = 2gh,$$

therefore $OH^2 = 2gh = v^2$, where v represents the velocity due to the head h.

Extracting the roots

$$OH = v.$$

The velocity v due to the head h for the case under consideration may now be calculated and marked off from OH to any suitable scale (say 20 feet to 1 inch), as shown by the thick line in the diagram.

Draw c, c_1, c_2 parallel to GH, OF, and FG respectively, and scale off their lengths to the same scale that was used for v.

Apply these corrected lengths to the diagram, and measure off w_1, w_2, u, and the investigation is complete.

The head h is the total head, less about 15 per cent. allowance for losses by friction.

According to the properties of triangles c_1^2 may be expressed

$$c_1^2 = c^2 + w_1^2 - 2cw_1 \sin a \quad - \quad - \quad (12)$$

substituting the values for w_1^2 and w_1 given by (11) and simplifying

$$c_1^2 = c^2\left\{1 + \left(\frac{r_1}{r_2}\right)^2 . \left(\frac{A}{A_2}\right)^2 \sin a_2^2 - 2\frac{r_1}{r_2} . \frac{A}{A} . \sin a . \sin a_2\right\} \quad (13)$$

but by (5*b*) $c_1^2 = c^2 . \left(\frac{A}{A_1}\right)^2$, therefore the quantity contained in the brackets in (13) equals $\left(\frac{A}{A_1}\right)^2$.

$$\left(\frac{A}{A_1}\right)^2 = \left\{1 + \left(\frac{r_1}{r_2}\right)^2 \cdot \left(\frac{A}{A_2}\right)^2 \sin \alpha_2{}^2 - 2\frac{r_1}{r_2}.\frac{A}{A_2}.\sin \alpha . \sin \alpha_2\right\} \quad (14)$$

substituting this value in equation (6)—

$$h = \frac{1}{2g}c^2\left\{1 + \left(\frac{A}{A_2}\right)^2 - \left[1 + \left(\frac{r_1}{r_2}\right)^2 \cdot \left(\frac{A}{A_2}\right)^2 \sin \alpha_2{}^2 - 2\frac{r_1}{r_2}.\frac{A}{A_2}.\sin \alpha . \sin \alpha_2\right]\right\} \quad (15)$$

simplifying and extracting the value of c—

$$c = \sqrt{2gh}.\sqrt{\frac{1}{\left(\frac{A}{A_2}\right)^2 \cdot \left\{1 - \left(\frac{r_1}{r_2}\right)^2 . \sin \alpha_2{}^2\right\} + 2\frac{r_1}{r_2}.\frac{A}{A_2}.\sin \alpha . \sin \alpha_2}} \quad (16)$$

$$c = K\sqrt{2gh}.$$

The values of c, c_1, c_2, w_1 and w_2, as investigated by the above-described methods, are the theoretical values, and do not take into account the losses caused by friction of the pipes and vanes. There are several separate causes for loss in reaction turbines, namely, friction of vertical supply pipe; friction of guide vanes; friction of wheel buckets; loss by leakage between guide vanes and top of wheel buckets; loss from energy represented by the velocity u; and when the wheel is not running at its best speed, loss by impact due to the angle of inflow α_1 being different to the corresponding angle of the wheel buckets.

The first of these losses may be calculated by the formula—

$$h_1 = f_o \frac{l}{d} . \frac{v^2}{2g} \quad - \quad - \quad - \quad (17)$$

By making the velocity v small, such as 3 to 5 feet per second, the head lost on this account is very small. Values of f_o have already been given in Chapter II.

The head lost by friction of guide vanes is given by the equation—

$$h_{11} = f . \frac{c^2}{2g} \quad - \quad - \quad - \quad (18)$$

in which f has the value .11, determined by experiment.

The losses occurring through leakage between the guide vanes and wheel are dependent on the pressure h_1 at that point, and on the width of opening between the guides and wheel, usually $\frac{1}{8}$ inch. As an attempt to calculate this loss would require a good many assumptions to be made, it is advisable to make an allowance as observed from good examples of turbines. The loss by leakage is found to be fairly represented by about 4 to 5 per cent. of the total head, so that we may write the equation—

$$h_{21} = .04\text{H to } .05\text{H} \qquad - \quad - \quad (19)$$

It may be here observed that on account of this leakage the velocity c will actually rise by 4 or 5 per cent. above what is required by the ratio $\frac{A}{A_2}$. In fact the gap between the guides and wheel is a sort of useless addition to the areas A_1 and A_2.

The losses occurring in the wheel buckets may be calculated by a modification of equation (18), for instead of a uniform velocity we have a velocity varying from c_1 to c_2. Assuming that the change from c_1 to c_2 takes place by uniform acceleration, then the equation becomes—

$$h_{31} = f.\frac{c_1^2 + c_2^2}{2}.\frac{1}{2g} \qquad - \quad - \quad - \quad (20)$$

in which f has the same value as before.

The energy lost in every pound of the off-flowing water due to the velocity u is represented by the equation—

$$h_{41} = \frac{u^2}{2g} \qquad - \quad - \quad - \quad (21)$$

By combining the above equations the useful head h_u, representing the proportion of the total head H, which is converted into useful work, may be found.

$$h_u = \text{H} - (h_1 + h_{11} + h_{21} + h_{31} + h_{41}) \qquad - \quad (22)$$

The head h for use in solving the equations (7) and (16) is given by the equation—

$$h = H - (h_1 + h_{11} + h_{21} + h_{31}) \quad - \quad - \quad (23)$$

consequently--

$$h_u = h - h_{41} \quad - \quad - \quad - \quad (23a)$$

The equations (17) to (23) are not in a form for direct use, as they contain the quantities c, c_1, c_2, u, which are unknown until equation (17) or (23) has been solved. They might, however, be worked out in suitable form and included in equations (7) and (16), but there is then the disadvantage that all operations are conducted at once, and it becomes difficult to follow the effect of the various losses.

A very near approximation to the value of c, corresponding to the head h, as given by equation (23), is obtained by first calculating c, c_1, c_2 by equation (7) or (16), and then using these values in solving equation (23). This method is, of course, not strictly correct; but as some of the quantities in the equations (17) to (20) are only approximate, it is useless to be too critical.

An example worked out for a head of 10 feet, and $A = A_2$, $a = a_2 = 60°$, as shown in Fig. 158, gives a loss of head of 1.66 feet, or

$$h = H - 1.66 = 10 - 1.66 = 8.33, \text{ or } 83.3 \text{ per cent.,}$$

or a loss of 16.66 per cent. due to friction. Experiments conducted on existing turbines give values of 82 to 86 per cent. for h. There is a further loss of about 12.6 per cent. due to the velocity u of off-flow, so that the useful head h_u given in equation (22) is—

$$\begin{aligned} h_u &= H - (1.66 + 1.26) \\ &= 10 - 2.92 = 7.08, \text{ or } 70.8 \text{ per cent.} \end{aligned}$$

Allowing 2.8 per cent. for shaft friction leaves 68 per cent. for the brake efficiency of the turbine, which is a good performance for the design under consideration. By increasing $a = a_2$ to 66° the brake efficiency is improved to

73 per cent., and by further enlargement of the angles a and a_2 a better efficiency may be expected.

The calculations for the velocities being completed, the proportions of the turbine, such as width of guide and wheel passages, diameter, number and depth of buckets and guide passages, may be settled. If the wheel is required to give a certain brake horse-power, the corresponding quantity of water, Q cubic feet, can be calculated from the value h_u in equation (22).

$$Q = \frac{\text{B.H.P.} \times 33000}{60 \times 62.27 \times h_u} \qquad (24)$$

The area A necessary to pass this quantity is $A = \frac{Q}{c}$, so that the width of guide passages e_1 may now be calculated from the formula—

$$e_1 = \frac{A}{2\pi r_1 \cos. a - z_1 t_1}$$

in which r_1 represents the mean radius for an axial wheel, and z_1 the number of guide vanes, and t_1 their thickness. As, however, the passage of the wheel vanes across the openings of the guide passages the effective area A is diminished below the value above found, so that if z_1 is the number of wheel vanes, and t_2 their thickness, the equation becomes—

$$e_1 = \frac{A}{2\pi r_1 \cos. a - (z_1 t_1 + z_2 t_2)} \qquad (25)$$

In the same way the width e_2 of the wheel at outflow is found by—

$$e_2 = \frac{A_2}{2\pi r_2 \cos. a_2 - z_2 t_2} \qquad (26)$$

If the diameter chosen gives unsuitable values for e and e_2, a new diameter must be selected, and the calculation repeated. The number z_1 of guide vanes may be found by making the width of opening of passage, measured at right angles to the direction of flow, equal to about 5 inches. Then

number z_2 is then given either equal to z_1 or slightly greater. The depth of the buckets may now be settled so as to give a change of direction to the water not too abrupt. The dimensions satisfying this condition varies from about 8 to 12 inches. The depth of guide passages may be from $\frac{3}{4}$ to 1 of the depth of the buckets.

Instead of discharging the water from the buckets direct into the tail-race, it is evident we may lift the wheel some

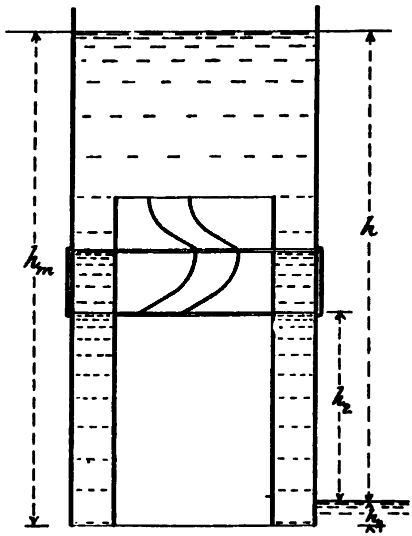


Fig. 162.

height above the tail-race level and connect a hermetically sealed pipe, called a suction tube, to the guide passage frame enclosing the wheel from the atmosphere, and having an opening under the tail-water. If this pipe be filled with water, the rate of flow in it, when the turbine is at work, will depend upon its sectional area A_3, and compared with c the velocity in this pipe will be $\frac{A}{A_3}c$, and making the outflow area

A_4, the velocity of off-flow becomes $\frac{A}{A_4}c$. If the velocity of off-flow c_4 is the same as u, as it will be if the suction tube is of annular form, as shown in Fig. 162, having a width e_2, then the calculation is the same as that for a turbine without suction tube. There will, of course, be greater loss owing to the increased wetted surface of the annular suction tube above that of a plain tube.

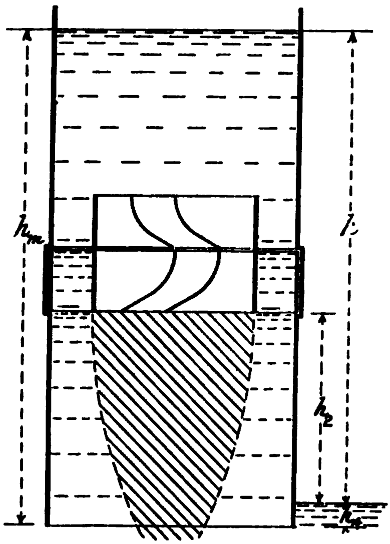


Fig. 163.

If, now, the inner ring be removed, the off-flow area A_4 appears to have the full area of the suction tube, but whether this is so or not depends on several circumstances. The water in flowing from the wheel with the velocity u cannot at once alter its velocity to $c_3 = c_4$ to suit the area of the suction tube $A_3 = A_4$. The result is that there is a central core of comparatively dead water somewhat of the shape shown in Fig. 163. As the water passes down the suction

tube the velocity changes from u by values approaching nearer and nearer to c_4, until, if the tube is sufficiently long in proportion to its diameter, the velocity c_4 is reached. If the tube is short and of large diameter, the velocity outflow c_4 will not be represented by $\frac{A}{A_4}c$, but will have some higher value, as the whole area A_4 is not in that case effective.

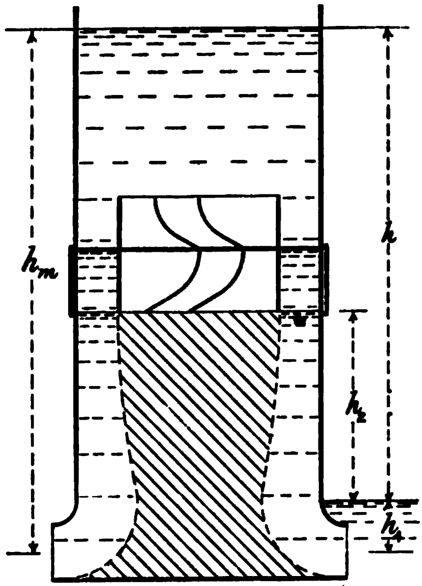


Fig. 164.

This uncertainty of the velocity c_4 may be overcome by putting a bottom plate into the suction tube and curving the lower edge outwards to form a lip as shown in Fig. 164. The area A_4 has now a definite value, and if an inner tube is to be used, it should be somewhat similar in shape to the dotted line in Fig. 164. Experiments on turbines prove that if this inner tube is only an approximation to the true

shape, the efficiency of the wheel falls below the efficiency without an inner tube. Generally it is advisable to leave out the inner tube.

By altering the area A_4 the velocity c_4 may be altered, but in altering it will have an effect on the other velocities c, c_1, c_2, so that a new equation must be evolved before c can be ascertained. Referring to Fig. 164, the useful head h is represented by $h_m - h_4$, so that as before –

$$h = h_m - h_4 = h_1 - h_4 + \frac{c^2}{2g} \qquad (27)$$

adding $h_2 - h_2$ to each side—

$$h = h_1 - h_2 + h_2 - h_4 + \frac{c^2}{2g} \qquad (28)$$

By equations (3) and (4) $h_1 - h_2$ has been shown equal to $\frac{c_2^2}{2g} - \frac{c_1^2}{2g}$, similarly—

$$h_2 + \frac{u^2}{2g} = h_4 + \frac{c_4^2}{2g} \qquad (29)$$

therefore—

$$h_2 - h_4 = \frac{c_4^2}{2g} - \frac{u^2}{2g} \qquad (30)$$

Substituting these values in equation (28)—

$$h = \frac{1}{2g}(c^2 + c_2^2 - c_1^2 + c_4^2 - u^2) \qquad (31)$$

Expressing c_1, c_2, u, c_4 in terms of c—

$$c_1 = \frac{A}{A_1}c, \; c_2 = \frac{A}{A_2}c, \; u = c_2 \cos. a_2 = \frac{A}{A_2} c \cos. a_2, \; c_4 = \frac{A}{A_4}c.$$

Substituting these values in (31)—

$$h = \frac{1}{2g}c^2\left\{1 + \left(\frac{A}{A_2}\right)^2 - \left(\frac{A}{A_1}\right)^2 + \left(\frac{A}{A_4}\right)^2 - \left(\frac{A}{A_2}\right)^2 \cos.^2 a_2\right\} \qquad (32)$$

The expression $1 + \left(\frac{A}{A_2}\right)^2 - \left(\frac{A}{A_1}\right)^2$ has already been dealt

with in equation (15), and $\left(\frac{A}{A_2}\right)^2 \cos.^2 a_2$ may be written $\left(\frac{A}{A_2}\right)^2 (1 - \sin^2 a_2)$, therefore—

$$h = \frac{1}{2g} c^2 \left\{ \left(\frac{A}{A_2}\right)^2 \sin^2 a_2 \left[1 - \left(\frac{r_1}{r_2}\right)^2\right] + 2\frac{r_1}{r_2} \cdot \frac{A}{A_2} \sin a \sin a_2 + \left(\frac{A}{A_4}\right)^2 \right\} \tag{33}$$

consequently—

$$c = \sqrt{2gh} \cdot \sqrt{\frac{1}{\left(\frac{A}{A_2}\right)^2 \sin^2 a_2 \left[1 - \left(\frac{r_1}{r_2}\right)^2\right] + 2\frac{r_1}{r_2} \cdot \frac{A}{A_2} \sin a \sin a_2 + \left(\frac{A}{A_4}\right)^2}} \tag{34}$$

$$= K\sqrt{2gh}.$$

The value of c having been found from the above equation, the corresponding values of c_1, c_2, w_1, w_2, u, c_3 and c_4 may be easily ascertained, and after making due allowance for frictional losses, the calculation of the dimensions of the wheel proceeded with as before.

As regards the losses, the frictional loss in the suction tube may be ascertained with sufficient accuracy by the equation—

$$h_1 = f_o \cdot \frac{l}{d} \cdot \frac{c_3^2}{2g},$$

in which c_3 represents the average velocity in the suction tube, combining this with equation (17)—

$$h_1 = f_o \cdot \frac{l}{d} \cdot \frac{v^2}{2g} + f\frac{l}{d} \cdot \frac{c_3^2}{2g} \quad - \quad - \quad (35)$$

Equation (21) must now be written—

$$h_{41} = \frac{c_4^2}{2g} \quad - \quad - \quad - \quad (36)$$

By substituting these new values in equations (22) and (23) the values of h_u and h are ascertained.

If the velocity c_4 is less than u, an examination of equation (23*a*) shows that h_u is greater for a suction tube turbine than for the same turbine without a suction tube.

The amount by which the head h_u is increased is represented by—

$$h_{2u} = \frac{1}{2g}(u^2 - c_4^2).$$

It has already been pointed out that the change of velocity from u to c_4 seldom takes place without loss from eddy currents. If the whole head h_{2u} represented by the change be lost, then h_u has the value—

$$h_u = h - \frac{u^2}{2g}$$

as before. Generally h_u will lie between these two extremes, and may be expressed—

$$h_u > h - \frac{u^2}{2g} < h - \frac{u^2}{2g} + \frac{c_4^2}{2g} \quad \cdot \quad (37)$$

No mention has been made as to the maximum height of suction tube allowable in a turbine. The height to which water will stand in a vacuum tube when balanced by the pressure of the air is 34 feet, so that with perfect conditions the height of the vane outlet above the tail-water level may be 34 feet. There are objections to this in practice, as any slight reduction in the velocity u would at once cause a vacuum in the suction tube, and on the velocity u again increasing, the two surfaces of water would again approach with a loud report known as the water hammer. Further than this, all water contains more or less air and other gases in solution, which are given off on a reduction of pressure. The gases thus entering the suction tube added to the air leaking in through the joins would very soon destroy the continuity of water with a con-

sequent loss of head. In practice the following heights of suction tube are found to be the limits:—

Diameter. Feet.	Height. Feet.	Diameter. Feet.	Height. Feet.
13.0	9.84	4.9	19.68
11.5	11.15	3.3	26.24
9.8	12.46	2.0	27.88
8.2	13.77	0.98	29.52
6.5	14.76	0.49	31.16

CHAPTER XXI.

DESIGN OF TURBINES IN DETAIL.

In the preceding chapters it has been pointed out that there is one particular speed at which any turbine is best suited to run. At this speed the efficiency is highest, and consequently the water consumption per horse-power lowest. If the work being performed by the turbine is for any reason reduced, the turbine will accelerate in speed, and if the work is increased the speed is reduced. In impulse turbines this alteration of the speed causes loss from impact of the water against the wheel buckets, and the consumption per horse-power is increased. In reaction turbines there is a loss by impact due to this change of speed, but in addition the velocity of flow through the wheel is liable to considerable variation. As the speed of the wheel is reduced, the velocity increases until, when the wheel ceases to revolve, the velocity approaches nearly to that due to the head acting on the wheel.

If, on the other hand, the speed of the wheel is increased above its best value, in most designs of reaction turbines the velocity of the water through the wheel is again increased. This is most noticeable in the designs illustrated by Figs. 158 and 159. In a design similar to Fig. 157 the increase would be trifling.

This increase of speed is due to the fact that while one part of the vane is acted upon by the water, the other part is acting upon the water and forcing it through the wheel after the manner of a centrifugal pump. This unsuitability of the turbine to a variation of speed is of very little practical importance, as the usual requirement is for a fixed speed

with variation of power. In most instances the water supply is none too plentiful, especially during the summer months, so that it becomes necessary to make the water consumption in some measure proportional to the power required. There are many forms of regulating devices for varying the power and water consumption of turbines. In some instances these appliances are worked by hand, while in others their action is rendered automatic through the agency of a governor. The most primitive form of regulator used for both impulse and reaction turbines consists of a sluice placed in the head-race. The part closing of this sluice causes the water on passing it to acquire a greater velocity and consequent loss of head. The turbine is now working with a considerably reduced head, but with the same velocity, hence the angle of the wheel vanes is unsuited to the new condition, and there is consequently considerable loss of efficiency and increased water consumption per horse-power. If the head is reduced to one-half, the efficiency would fall to about 30 per cent. This form is evidently unsuitable.

A form of regulator, which is applicable to reaction turbines only, consists of a sluice applied to the tail-race or suction tube. Referring to Fig. 164, the bottom plate of the suction tube might be made adjustable in position, so that by rising or falling it would alter the area A_4 of off-flow. Instead of adjusting the bottom plate, it is usual to cause the lower part of the suction tube to telescope on the upper part, thus altering the area A_4. Though this form of regulator does not alter the efficiency of the turbine much when the variation of the opening is slight, still it has very little effect on the quantity of water flowing even when closed to one-fourth of its full area. This fact is easily accounted for, as the reduction of area A_4 causes less difference between the pressures h_m and h_4, and a consequent reduction of useful head h. The remainder of the total head is absorbed in giving to the water a high velocity of off-flow c_4.

Fig. 165 shows a form of regulator which has been used at times. Between the guide vanes A and wheel vanes B a sliding regulator C is placed, having holes corresponding with the openings in the guides. On causing the regulator C to pass across the faces of the guide openings, their area is reduced, and the flow of water consequently interfered with. An inspection of the drawing will show that this form is very unsuitable for reaction turbines, as there is a sudden enlargement of area on entry to the wheel. For impulse turbines the effect on the efficiency is not very detrimental within limits, but owing to the distance between the guide openings, the turbine requires to be larger in size for a given power, thus increasing the cost.

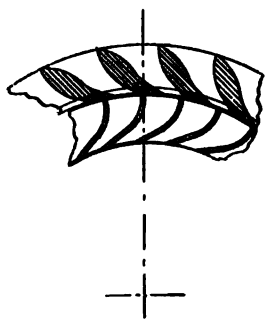


Fig. 165.

A modification of the above-described regulator has already been illustrated in Fig. 147, as applied to a Girard or partial admission impulse turbine. In this form the sliding regulator has no holes, but advances from one side, thereby cutting out entirely one or more of the guide passages. In the figure all the passages are shown closed. This method of regulation gives very good efficiencies, and has been applied in various forms to both impulse and reaction turbines.

One modification of the above-described regulator is shown in Fig. 166, where each guide passage is provided with a vertically operated sluice A, having the appearance of a spade. This sluice is so operated that the guide passage is either fully open or closed. The sluices are controlled by a circular guide rail which may be revolved by hand or by a governor mechanism. This guide rail consists of an upper and lower rail B and C, communicating

at two places by means of sloping grooves or cams, such as D. Small rollers attached to the sluices bear on the guide rails B when the guide passages are open; and on revolving the circular guide rail these rollers pass in turn down the sloping grooves D on to the lower guide rail C, thus causing the sluices A to enter the guide passage and stop the flow of water.

Owing to the length required for the efficient working

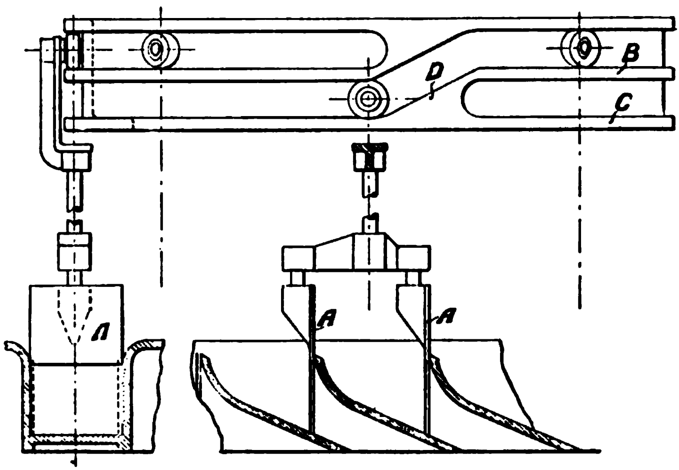


Fig. 166.

of the sloping groove it is usual to attach several sluices to one roller. By having two sloping grooves placed opposite to each other the closing of the guide passages always takes place equally on two opposite sides of the wheel, so that the wheel is always truly balanced about its centre, with a consequent minimum of friction. When this system of regulating is applied to reaction turbines it becomes necessary, especially if the velocity of flow through the wheel buckets is great, to make some provision

for the velocity of the water being reduced gradually, and shock thereby avoided. The space between the guide vanes and wheel may be used for this purpose, for as the wheel vanes pass under the guide passages containing dead water, the pressure will be reduced, and the continued flow of the water in the wheel buckets will cause air to be sucked in through this space. On the wheel buckets again coming under the active guide passages this air is expelled in front of the entering water with little or no loss of efficiency. When the buckets are large, and this method of inlet would prove insufficient, air ports in the guide passages are arranged

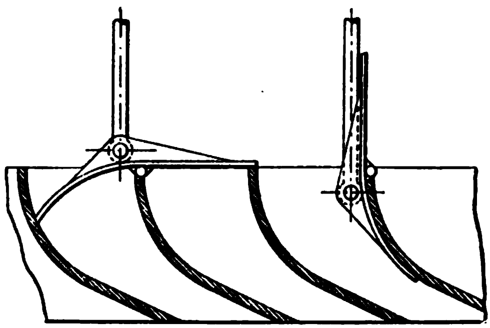


Fig. 167.

to open automatically on the shutting of the sluice. Of course this method of air-cushioning is not allowable in suction tube turbines.

Instead of only two sloping grooves there is sometimes provided a groove for each roller; the regulation then ensues from the whole of the guide passages being more or less opened or closed. The efficiency of this form is very low, and the method cannot be recommended.

Fig. 167 shows a regulator in which hinged flaps, attached to the top edge of alternate guide vanes, are used. These flaps are caused to oscillate through about 90° of angle,

thus opening or closing two guide passages. The motion of the flap is brought about by reciprocation of the vertical rod A, operated by a cam groove similar to that shown at D, Fig. 166.

A very well known regulator is illustrated in Fig. 168,

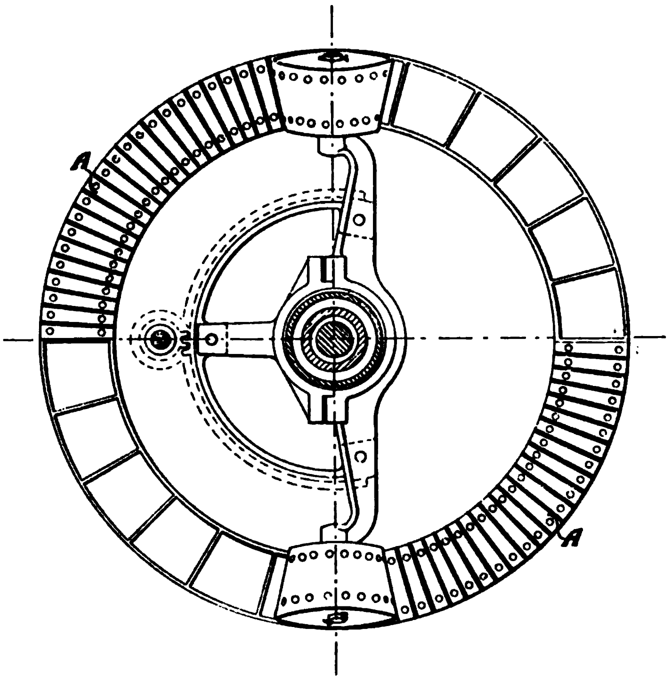


Fig. 168.

known as the scroll regulator. A scroll or blind A, made of leather or indiarubber, stiffened with metal strips, is attached by one of its ends to the top of the guide apparatus. The other end is attached to a roller attached to the end of a revolving arm, and on the arm being revolved the blind

is wound upon the roller or *vice versâ*. Two of these scrolls and rollers are usually applied to a turbine, so that the wheel is always balanced. In a modified form of this regulator each scroll is displaced by a half ring of metal. The guide passages of one-half of the turbine are deflected upwards while the other half are deflected downwards, as seen in Fig. 169. Each of the two half rings A B is thus

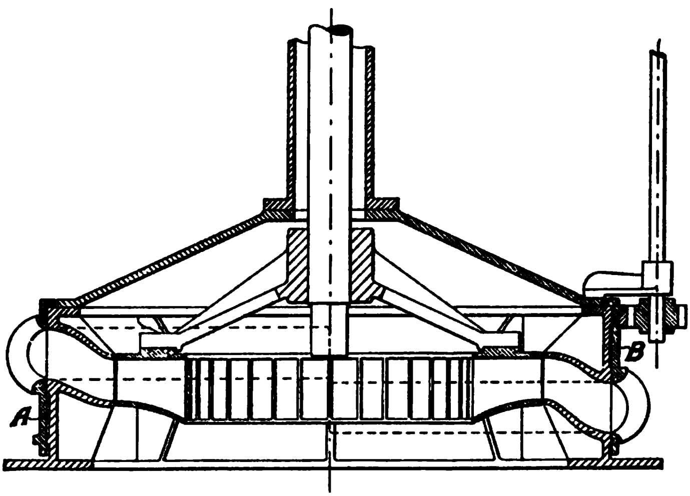


Fig. 169.

enabled to slide off the passages it has to control, without covering up the passages under the control of the other.

The arrangement shown in Fig. 170 was used by Professor James Thomson in his inward flow turbines. Some of the guide vanes are hinged, so that on being oscillated they reduce the opening of the guide passages. The movement is effected by link-work operated from a hand-wheel or governor mechanism.

A turbine has been designed by Nagel and Kämp in

which both the guide passages and wheel buckets are reduced in area. The arrangement is as follows:—The guide apparatus is provided with a false or movable side having a slot for each guide vane to pass through, and on this side being advanced towards the other side of the guide apparatus in a radial wheel the effective area is reduced. The wheel is arranged to rise or fall on its axis, and in

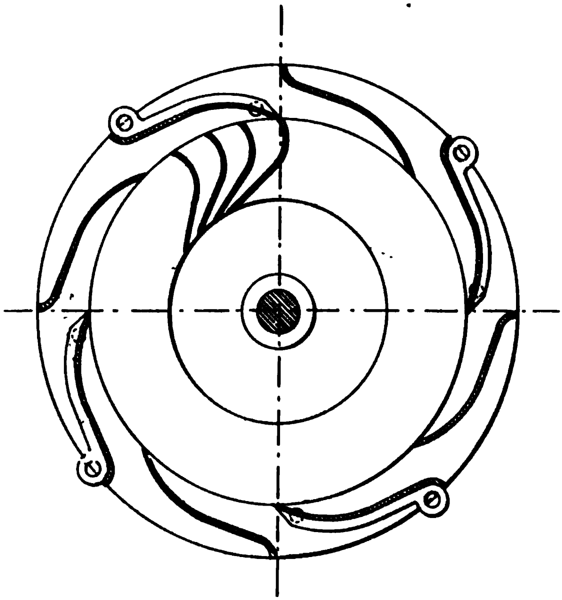


Fig. 170.

so doing operates this false side. One side of the wheel buckets is also movable in relation to the rest of the wheel, but fixed in relation to the guide vanes. Thus on the wheel being moved on its axle, the guide and wheel passages are altered in area. The arrangement has the disadvantage that it is costly and complicated.

The only other form of regulator requiring notice consists

of a ring capable of sliding over the outflow openings of the wheel buckets. Fig. 171 shows this arrangement as applied to the large turbines at Niagara Falls. The wheel buckets A and guide passages B are divided by two partitions so as practically to form three turbines side by side, and as the wheel is already double, having a top and bottom turbine, there are really six turbines coupled to one shaft. There are two rings, such as C, connected together by rods passing through guides, and as these rings are advanced over the

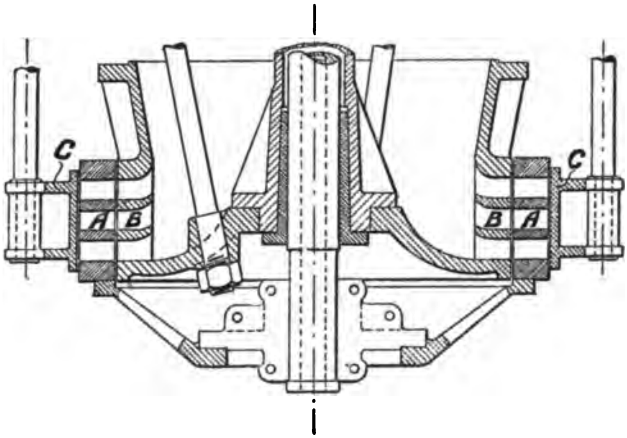


Fig. 171.

outflow areas, two of the six turbines are throttled, the remaining four still performing their full duty without loss of efficiency in them. When the two wheels are entirely closed the power is reduced to about two-thirds, and when four are closed the power is slightly under one-third of the full power. This form is particularly suited to large wheels, and those working with a large head of water and correspondingly high velocity of flow. The figure shows the lower portion only of a Niagara turbine.

EXAMPLES.

Impulse turbines (Example 1).

A quantity of 10 cubic feet of water per second is available, and after deducting head lost in pipe friction and bends, it is estimated that the available head is 332 feet.

$Q = 10$
$H = 332$

$$\text{Maximum H.P.} = \frac{Q \times 62 \times 60 \times H}{33000} = 374$$

Select $\alpha = 74^\circ$
" $\alpha_2 = 78^\circ$

$c = .95\sqrt{2gh} = 138.4$ feet per second $(h = H - h_0)$.

Owing to the small quantity of water and high velocity, the turbine must be an outward flow Girard wheel with partial admission.

Any number of revolutions per minute may now be selected.

Select R = about 210 per min. = 3.5 per sec.

In Fig. 144, w_1 is much less than c_1, take $w_1 = 63$ for purposes of trial, then

$$\frac{w_1}{\text{R per second}} = \text{circumference, or } \frac{60}{3.5} = 17 \text{ ft. or } 5.4 \text{ ft. dia.}$$

Select depth h_0 of vanes = 6 inches, then outer diameter = 6.4 feet. Draw out diagram, as Fig. 144, to scale, making

$\alpha = 74^\circ$ $\quad c = 138.4$ $\quad r = 2.7$
$\alpha_2 = 78^\circ$ $\quad w_1 = 63$ $\quad r = 3.2$

(Suitable scales for c and r are 50 feet = 1 inch and 1 foot = 1 inch respectively), then by measurement $c_1 = 80$, $w_2 = 74$, $c_2 = 76$, $u = 15$, $\alpha_1 = 61^\circ$; by calculation (equation 3, p. 275), $c_2 = 75.8$.

As the values of c_2 by the two methods practically agree, the assumption $w_1 = 63$ was correct. If these values had differed widely, the mean between the two should then be

taken, and the other values, c_1, etc., found by correcting the diagram.

The brake H.P. may now be found.

c = 138.4 is velocity due to 300 ft. head	∴ head lost =	32	=	10	%
c_2 = 76 „ „ 85.5 „	} „	= 19.5	=	5.8	%
c_1 = 80 „ „ 105.0 „					
u = 15 „ „ 4 „	„	= 4	=	1.2	%
				17.0	%
Shaft friction				5	%
				22	%

Then 100 : 374 :: 78 : B.H.P.

B.H.P. = 291.7, or say 280 H.P. available.

The dimensions of guides and buckets may now be settled.

$$A = \frac{Q}{.89c} = \frac{10}{.89 \times 138.4} = .081 \text{ square feet.}$$

This is equivalent to an opening 3 inches wide × 4 inches measured at right angles to direction of flow.

e_1 for guides = 3 inches, and using two guide passages, their opening will be 2 inches each, or $4\frac{3}{4}$ inches measured on circumference. The pitch of wheel buckets must not be less than 5 inches, and their thickness is $\frac{1}{4}$ inch.

$$\text{Number of wheel buckets} = \frac{5.4 \times 3.14 \times 12}{5\frac{1}{4}} = 38.$$

By scaling the width of inlet to bucket at right angles to c_1, and outlet at right angles to c_2, their values are $3\frac{1}{8}$ and $2\frac{1}{8}$ inches respectively—

$e_1 \times 3\frac{1}{8} \times c_1 = e_2 \times 2\frac{1}{8} \times c_2$ ∴ $e_2 = 4.9$ inches, say $5\frac{1}{4}$ inches to give extra clearance.

Collecting the values—

Q = 10 ft.	$a = 74°$	$c = 138.4$ ft.	e_1	= 3 in.
H = 332 „	$a_1 = 61°$	$c_1 = 80$ „	e_1 (for buckets)	= $3\frac{1}{4}$ in.
H.P. = 374	$a_2 = 78°$	$c_2 = 76$ „	e_2	= $5\frac{1}{4}$ „
B.H.P. = 291	$w_1 = 63$ ft.	$r_1 = 5$ ft. 5 in.	z_1	= 2
Load = 280 H.P.	$w_2 = 74$ „	$r_2 = 6$ „ 5 „	z_2	= 38

Regulation as shown in Fig. 147.

Example 2—

Quantity of water - - $Q = 30$ feet per sec.
Head - - - - - $H = 15.6$ feet.
$h = H - h_0 = 14.5$ „

$$\text{Max. H.P.} = \frac{Q \times 62.2 \times 60 \times H}{33000} = 52.8.$$

Select $\alpha = 68^\circ$
„ $\alpha_2 = 72^\circ$

$c = .95 \sqrt{2gh} = 28.88$

$A = \dfrac{Q}{.89c} = 1.17$ square feet.

Select mean diameter of wheel for axial flow, 3 feet.

As h_0 is a large percentage of the total head, c_2 may be taken approximately equal to c_1.

Draw out the diagram, Fig. 142, making u vertical, and select $w_1 = w_2 = 15\frac{1}{2}$ for trial. Then for $c = 28.88$, $c_1 = 15.5$, and $c_2 = 16.5$ by measurement.

By equation (3), p. 275, $c_2 = 16.5$.

The correct values are therefore $c_1 = 15.5$, $c_2 = 16.5$, $w_1 = w_2 = 15.5$, $u = 6$.

The brake H.P. may be found—

$c = 28.88$	velocity due to		13 ft.	∴ head lost 1.5 ft.	$= 9.7$ %
$c_1 = 15.5$	„		3.75 „	„ .5 „	$= 3.2$ %
$c_2 = 16.5$	„	$4.25 - h_0$	$= 3.25$ „		
$u = 6$	„		.56 „	„ .56 „	$= 3.25$ %
					16.15 %
Shaft friction					4.00 %
					20.15 %

Then 100 : 52.8 :: 79.8 : B.H.P.
B.H.P. = 42, or say 40 H.P. available.

By equation (4), p. 276, e_1 may be found. Take $z_1 = 20$, $t_1 = \frac{1}{4}$ inch $= .02$ feet.

$$e_1 = \frac{A}{2\pi r \cos.\alpha - z_1 t_1} = \frac{1.17}{2 \times 3.14 \times 1.5 \times .374 - .4} = .37 \text{ ft.} = 4\tfrac{1}{2}''$$

$A_2 = \dfrac{Q}{c_2} = \dfrac{30}{16.5} = 1.8$. Take $z_2 = 18$, $t_2 = \frac{1}{4}$ inch $= .02$ ft.

Then $$e_2 = \frac{A}{2\pi r_2 \cos.\alpha_2 - z_2 t_2} = \frac{1.8}{2 \times 3.14 \times 1.5 \times .309 - .36}$$

$= .7$ feet $= 8\frac{1}{2}$ inches nearly.

Allowing for clearance, make $e_2 = 9$ inches.

$$\text{Revolutions per min.} = \frac{w_1}{3.14 \times 3} \times 60 = 98.$$

Collecting the values—

Q $= 30$ ft.	$\alpha = 68°$	$c = 28.88$ ft.	e_1	$= 4\frac{1}{2}$ in.
H $= 15.6$,,	$\alpha_1 = 46°$	$c_1 = 15.5$,,	e_1(for buckets)	$= 4\frac{3}{4}$,,
H.P. $= 52.8$	$\alpha_2 = 72°$	$c_2 = 16.5$,,	e_2	$= 9$,,
B.H.P. $= 42$	$w_1 = 15.5$ ft.	$r_1 = 1.5$,,	z_1	$= 20$
Load $= 40$ H.P.	$w_2 = 15.5$,,	$r_2 = 1.5$,,	z_2	$= 18$

Depth of buckets may be made 8 inches, and depth of guides 6 inches.

Regulation by any of the methods shown in Figs. 166, 167, 168, 169.

Reaction turbines (Example 3).

Quantity of water - - $Q = 30$ feet per sec.
Head - - - - $H = 15.6$ feet.

Jonval type $A = A_2$.
Max. H.P. $= 52.8$
Select $\alpha = 72°$
,, $\alpha_2 = 72°$

Draw out the diagram, Fig. 158, to any scale, making $c_2 = \dfrac{A}{A_2} c$. With the dimensions thus found for c, c_1, c_2, construct the diagram, Fig. 161.

Assume a loss of head of 15 °/ₒ, and find velocity due to $h = .85H$; mark off this value on the diagram to a suitable

scale, and complete the diagram and scale off values for c, c_1, c_2.

$c = 21.2$ feet.

$c_1 = 6.1$ „

$c_2 = 21.2$ „

w_1, w_2, and u may be found by graphic method from c_2—

$w_1 = w_2 = 20.5$

$u = 6.1$

Find brake H.P. By eq. (23 a), $h_u = h - h_{41} = .85\text{H} - \dfrac{u^2}{2g}$

$$= (85 - 04)\text{H} = .81\text{H}.$$

∴ Hydraulic losses - - - -	19 °/。
Shaft friction - - - - -	4 °/。
	23 °/。

Then 100 : 77 :: 52.8 : B.H.P.

B.H.P. = 40.65.

The dimensions of the wheel may now be settled.

$$\text{A} = \frac{\text{Q}}{c} = \frac{30}{21.2} = 1.41 \text{ square feet.}$$

Select radius 1.25 feet and $z_1 = 15$, $t_1 = \frac{1}{4}$ inch $= .02$ feet, $z_2 = 16$, $t_2 = .02$, then

$$e_1 = \frac{1.4}{2 \times 3.14 \times 1.5 \times .309 - (15 \times .02 + 16 \times .02)} \text{ by eq. (25)}$$

$$= .609 \text{ feet} = 7\tfrac{3}{8} \text{ inches.}$$

For Jonval turbines $e_2 = e_1$. Owing to the obstructing action of the vanes, A_2 is not in practice equal to A, but slightly greater.

$$\text{Revolutions per min.} = \frac{w_1}{3.14 \times 3} \times 60 = 130.$$

Collecting the values—

Q $= 30$ ft.	$a = 72°$	$c = 28.88$ ft.	$e_1 = 7\frac{3}{8}$ in.
H $= 15.6$ „	$a_1 = 0°$	$c_1 = 15.5$ „	$e_2 = 7\frac{3}{8}$ „
H.P. $= 52.8$	$a_2 = 72°$	$c_2 = 16.5$ „	$z_1 = 15$
B.H.P. $= 40.65$	$w_1 = 20.5$	$r_1 = 1.5$ „	$z_2 = 16$
Load $= 40$ H.P.	$w_2 = 20.5$	$r_2 = 1.5$ „	$t_1 = t_2 = \frac{1}{4}$ in.

Depth of buckets 6 inches and guides 6 inches.

Regulation by any of the methods shown in Figs. 166, 167, 168.

Example 4—

Design an inward flow radial turbine for the same conditions.

$Q = 30$ feet.

$H = 15.6$ feet. Max. H.P. $= 52.8$.

Select $\frac{A}{A_2} = \frac{1}{1.5} = .666$.

„ $a = 75°$.

„ $a_2 = 72°$.

„ $r_1 = 2$ feet.

„ $r_2 = 1.5$ feet.

Draw out the diagram as in Fig. 157 to any scale, making $c_2 = \frac{A}{A_2} c$, and observing ratio $r_1 : r_2$.

With the dimensions thus formed for c, c_1, c_2, construct diagram, Fig. 161.

Assume a loss of head of 15 %, and find velocity due to $h = .85H$, and mark off this value on the diagram to a suitable scale, and complete the diagram and scale off values for c, c_1, c_2.

$c = 25$ feet.

$c_1 = 7$ „

$c_2 = 16.8$ „

w_1, w_2, u may now be found by graphic methods from c_2—

$w_1 = 21.3$

$w_2 = 16.1$

$u = 5.3$

a_1 by measurement $= 24°$.

Find brake H.P. By equation (23a) $h_u = h - h_{41} = .85H - \frac{u^2}{2g}$

$= (.85 - .03)H = .82H$ $\therefore$

Hydraulic losses	-	18 %
Shaft friction -	-	4 %
		22 %

Then 100 : 78 :: 52.8 : B.H.P.

B.H.P. $= 41.18$. Say 40.

The dimensions of the wheel may now be settled.

$$A = \frac{Q}{c} = \frac{30}{25} = 1.2 \text{ square feet.}$$

Select $z_1 = 18$, $z_2 = 20$, $t_1 = t_2 = .02$ feet.
Then (by equation 25)—

$$e_1 = \frac{1.2}{2 \times 3.14 \times 2 \times .258 - (18 - .02 + 20 \times .02)} = .45 \text{ ft.} = 5\tfrac{3}{8}''$$

$$A_2 = \frac{Q}{c_2} = \frac{30}{16.8} = 1.8 \text{ square feet.}$$

Then (by equation 26)—

$$e_2 = \frac{1.8}{2 \times 3.14 \times 1.5 \times .309 - (20 \times .02)} = .73 \text{ ft.} = 8\tfrac{3}{4}''$$

$$\text{Revolutions per min.} = \frac{w_1}{3.14 \times 4} \times 60 = 102.$$

Collecting the values—

Q $= 30$ ft.	$a = 75°$	$c = 25$ ft.	$e_1 = 5\frac{3}{8}$ in.
H $= 15.6$,,	$a_1 = 24°$	$c_1 = 7$,,	$e_2 = 8\frac{3}{4}$,,
H.P. $= 52.8$	$a_2 = 72°$	$c_2 = 16.8$,,	$z_1 = 18$
B.H.P. $= 41.18$	$w_1 = 21.3$ ft.	$r_1 = 2$,,	$z_2 = 20$
Load $= 40$ H.P.	$w_2 = 16.1$,,	$r_2 = 1.5$,,	$t_1 = t_2 = \frac{1}{4}$ in.

Depth of buckets 6 inches, and guides 6 inches.
Regulation by any of the methods shown in Figs. 169, 170.

CHAPTER XXII.

WATER WHEELS.

IN the overshot water wheel, as illustrated in Fig. 172, the water acts upon the buckets or paddles of the wheels chiefly by its weight, passing from a trough or stream over the upper face of the wheel so as to fall against the surface of the buckets. This type of wheel is useful for low falls varying from 10 to 65 feet, the head water level not varying more than 2 feet. The efficiency varies from 60 per cent. to 75 per cent. The efficiency of the wheel is decreased by the loss of water, which arises owing to the horizontal velocity of the water when falling upon the wheel, and further loss results owing to the fact that the tail water does not flow freely from the wheel pit, but gives instead a certain amount of back wash in an opposite direction to the flow of the tail-race. The useful horse-power obtainable, assuming an efficiency of 65 per cent., is

$.65 \times \frac{62.25 \times QH}{550} = .074QH$, where H is the available head measured in feet.

The water should have a velocity greater than the circumference of the wheel; thus if the wheel has a peripheral velocity of 6 feet per second, the water should be flowing at about 10 feet per second. This velocity is obtained by falling through a height $10^2 = 2gh$ or $h = \frac{100}{64.4} = 1.55$ feet, or the water should enter the wheel at a position 1.55 feet below the surface level of the head water.

To remedy the practical losses arising from the non-clearance of the tail water, and to enable the wheel to be

immersed beyond the 1 foot extreme limit of immersion for an overshot wheel, the breast wheel, as shown in Fig. 173, is employed.

The water acts by weight only, dropping almost vertically into the buckets through the openings in the pen trough,

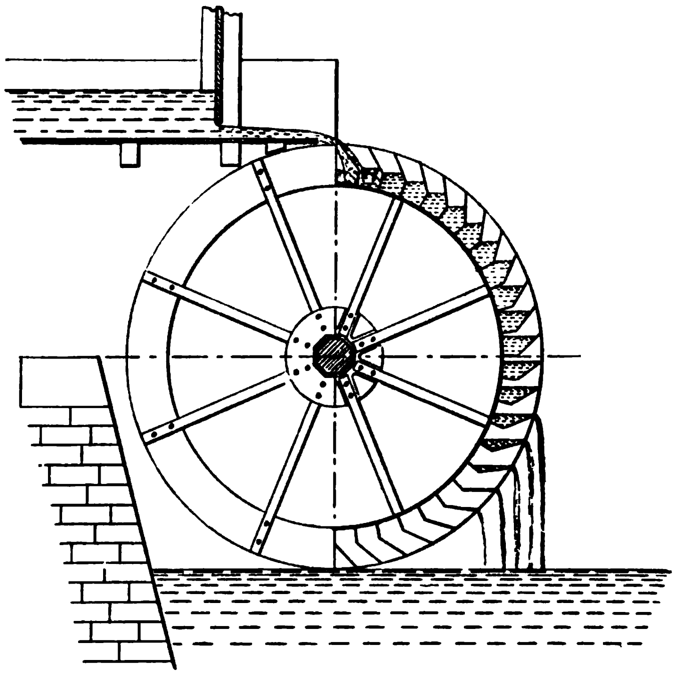


Fig. 172.

which is shaped to the circumference of the wheel. The masonry breast or curved edge adjacent to the wheel is not employed in the large wheels of this type where the diameter exceeds 19 feet. The buckets are only partially filled, and the space between the inner edges of the buckets and the wheel shrouding admits of free ventilation during

the movement of the wheel. The efficiency of the ordinary breast wheel varies from 70 to 75 per cent.

The oldest type of wheels known is that of the undershot, as illustrated in Fig. 174. The efforts of Poncelet led to great improvements in the efficiency of this wheel. It is used for falls up to 6 feet, acting on the same principle as

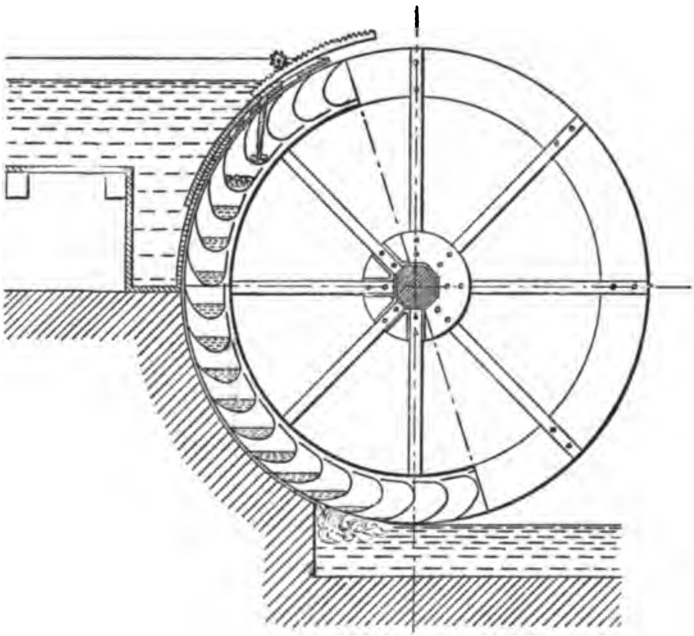


Fig. 173.

the impulse turbine. The stream of water should flow down an incline of 1 in 10 to impinge upon the curved blades near the bottom of the wheel and leave them with very little velocity and consequent work unabsorbed.

The diameter of the wheel should be at least twice the fall, the speed of the periphery between 50 and 60 per cent. of the velocity due to the fall measured to the centre of the inlet

orifice. The depth of the bucket in the radial direction equal at least to one-half the fall. The number of buckets found to be most efficient is 1.6, the diameter of the wheel in feet + 16.

Thus with a fall of 3 feet, and for a wheel of 12 horse-power, the diameter will be 6 feet.

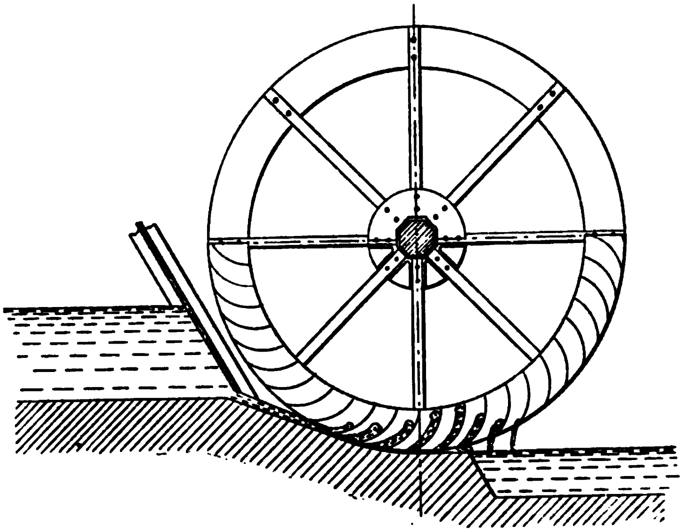


Fig. 174.

The fall being 3 feet, the velocity due to that height—

$$v = 8\sqrt{3} = 13.86 \text{ feet per second.}$$

$$\text{Number of revolutions} = \frac{6.93 \times 60}{\pi 6} = 22.$$

Assuming a duty of 60 per cent., then the cubic feet of water Q required will be—

$$Q = \frac{12 \text{H.P.} \times 530.5}{3 \times .6} = 3536,$$

which will require a width of 10 feet 6 inches with a depth of stream taken at 7 inches, and a discharge assumed as about 70 per cent. of the theoretical quantity on to the wheel.

CHAPTER XXIII.

HYDRAULIC ENGINES.

UNDER the head of hydraulic engines we propose to discuss the motors in which the hydraulic pressure, acting on a reciprocating piston in a cylinder, causes the revolution of a shaft from which power may be taken for doing work of any kind. Before discussing the best known types in detail, it is advisable to inquire into the causes of loss and means of prevention with a view to the production of an ideal motor.

The water pressure available may either be expressed in feet of head or as pressure per square inch, usually the latter for the type of motor under discussion. Whichever form is given, the conversion to the other is very simple. Since a column of water 1 square inch in section and 1 foot high weighs .434 lbs., it is evident that the pressure per square inch in pounds $\div .434$ will give the corresponding head in feet, or $\frac{p}{\cdot 434} = H$, and conversely $H \times .434 = p$. The head H or pressure p being known, the total energy per pound of water can be calculated.

According to the hydrodynamic equation the total energy of 1 lb. of water is—

$$E = \frac{p}{\cdot 434} + \frac{v^2}{2g} + \frac{L}{\cdot 434} \qquad (1)$$

where p represents the actual pressure in pounds per square inch at any point, and v the velocity of flow at the same point; while L represents the pressure in pounds per square inch lost from all causes between the source of supply and

the point under consideration. By assuming the source of supply to be close to the cylinder of the hydraulic engine, the quantity L may be taken as equal to o, as by so doing the investigation will be much simplified. We have now two quantities to deal with, namely, $\frac{p}{\cdot 434}$, representing the head producing the pressure p in pounds per square inch of the water passing into the cylinder; and $\frac{v^2}{2g}$ representing the energy absorbed in producing the velocity v of the flow into the cylinder.

We have already (Chapter II.) examined the conditions necessary for the change of velocity of the water without loss of energy on entering the cylinder. In a perfect design of motor the passage from the valve to the cylinder would require to be conical or trumpet-mouthed, allowing a change of velocity to occur without loss by eddy currents. Having arranged for the economical entry of the water to the cylinder, we are now at liberty to examine its action upon the piston, and as the water entering the cylinder must have a velocity corresponding to that of the piston, we must first investigate the true velocity of the piston at each part of its stroke.

In the accompanying diagram, Fig. 175, let A B C D represent the crank path of a hydraulic engine, then A C will represent the length of the piston stroke. When the crank-pin is on the dead centre A the piston has no velocity, whereas when the crank-pin arrives at B, assuming the connecting rod of infinite length, the forward velocity v of the piston is equal to the circumferential velocity v_c of the crank-pin; for all intermediate positions of the crank-pin between A and B the piston will have a series of velocities varying between o and v_c. The values of v may be expressed as a function of v_c. Take any point E in the crank path, and draw a tangent E G to represent the velocity v_c, and resolve the velocity v_c into its components E H, H G, in

which E H represents the horizontal velocity v of the piston corresponding to the point E. Let fall the vertical E F, then the triangle O E F is similar to the triangle of velocities E G H, so that

$$v : v_c :: \mathrm{EH} : \mathrm{EG} :: \mathrm{EF} : \mathrm{EO}$$

or

$$\frac{v}{v_c} = \frac{\mathrm{EF}}{\mathrm{EO}} = \sin\theta,$$

therefore

$$v = v_c \sin\theta \quad - \quad - \quad - \quad (2)$$

The velocity v of the piston, therefore, varies as the curve of sines, and its value at any point of the stroke may be found by the aid of a table of sines, or by describing a semicircle with radius $\mathrm{OB} = v_c$ to any suitable scale, when the ordinates, such as E F, $\mathrm{E}^1\ \mathrm{F}^1$, measured to the same scale, will give the velocity v for the corresponding points F F^1 of the stroke.

Having obtained the values of v, we can now by the aid of the hydrodynamic equation (1) find the corresponding values of p, as E may be substituted by $\frac{p_o}{\cdot 434}$; p_o being the pressure per square inch when the water is at rest. The equation then becomes—

$$\mathrm{E} = \frac{p_o}{\cdot 434} = \frac{p}{\cdot 434} + \frac{v^2}{2g} \quad - \quad - \quad (3)$$

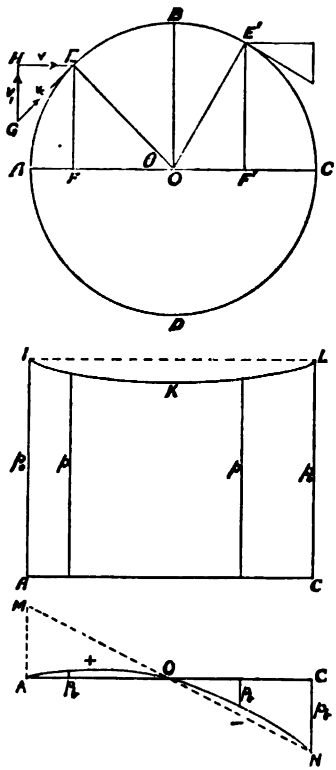
or

$$p = p_o - \frac{v^2}{2g} \times \cdot 434 \quad - \quad - \quad (3a)$$

The values of p thus obtained may be plotted as ordinates (Fig. 176). The curve thus produced will dip from the commencement of the stroke to the centre, when, owing to the decreasing velocity, it will again rise by a similar contour until at the end of the stroke it has the value p_o as at the commencement.

If the piston had moved forward through the stroke with a very small velocity, the pressure p_o would have remained constant throughout the stroke, so that the work done per

square inch of piston area would be p_0S foot-pounds, S being the length of stroke in feet. But p_0S represents the area of the parallelogram A I L C, which therefore is a measure of the total available energy per square inch of piston area.



Figs. 175, 176, and 177.

Instead, however, of the pressure p_0 we have the varying pressure p, so that the area of the diagram A I K L C represents the work done per square inch of piston area by the varying pressure p. The difference I K L between the dia-

grams A I L C and A I K L C, therefore, represents the kinetic energy due to the varying velocity of the water, and is a function of the quantity $\frac{v^2}{2g}$ in the hydrodynamic equation.

The diagram A I K L C gives the curve of pressures acting upon the piston, supposing that the base of the piston is always close up to the valve opening without any intervening water; this, however, is not the case, because as the piston recedes there is an increasing quantity of water to be accelerated. We will now proceed to deduce the curve of pressures necessary to produce this acceleration. It is well known in connection with steam engines that the force p_a necessary to produce acceleration of the reciprocating parts is represented by the equation—

$$p_a = \frac{w}{g} \cdot \frac{v_c^2(R - x)}{R^2} \qquad - \quad - \quad (4)$$

in which R is the radius of crank circle, x the distance travelled by the piston at any moment, and v_c the crank-pin velocity, all expressed in feet. Since $\frac{R - x}{R^2}$ at the commencement and termination of the stroke has the values $\frac{1}{R}$ and $-\frac{1}{R}$, the equation becomes—

$$p_a = \pm \frac{w}{g} \cdot \frac{v_c^2}{R} \qquad - \quad - \quad - \quad (5)$$

for those points, the + sign representing accelerating force, and the − sign retarding force.

Now the column of water to be accelerated may be considered as a piston, and we may assume for the time that its weight w per square inch of piston area is that of a column of water 1 square inch in area, and having a length equal to the length of the piston stroke. If the calculation be made, we get the straight line curve M N (Fig. 177), of which the vertical ordinates, such as A M, C N, represent the accelerating or retarding force p_a at the corresponding point of the stroke

for the assumed weight w. But as the only point of the stroke at which w really represents the weight of the water is at the end B, it is evident that the corresponding value of p_a is equal to the true value p_b for the varying water column. At the commencement of the stroke, as $w = 0$, so $p_b = 0$, and the values of p_b for any intermediate point may be found by multiplying the value of p_a given by equations (4) and (5) by the ratio $\frac{x}{2R}$, or—

$$p_b = \frac{w}{g} \cdot \frac{v_c^2(R - x)}{R^2} \cdot \frac{x}{2R} \quad \text{- - (6)}$$

The operation may, however, be more easily performed by graphic method on the diagram (Fig. 177) by dividing A C and C N into a similar number of equal parts, and drawing radial lines from O to C N and A M, and ordinates from A C, when points of intersection will be points on the curve by the principle of similar triangles. A O N in the figure represents the curve thus produced, and the ordinates p_b must be subtracted (observing the signs + and −) from the ordinates p_o in Fig. 176; thus we obtain the ordinates p_t in Fig. 178, which may be expressed by the equation (3*a* and 6 combined)—

$$p_t = p_o - \frac{v^2}{2g} \times .434 - \left\{ \pm \frac{w}{g} \cdot \frac{v_c^2(R - x)}{R^2} \cdot \frac{x}{2R} \right\} \quad \text{- (7)}$$

The diagram A I K P C thus produced is the diagram of work for the outward stroke of a hydraulic engine satisfying the condition laid down, namely, absence of hydraulic losses from friction between the source of supply and the cylinder. This diagram must accordingly have an area equal to the area of the parallelogram A I L C B, and consequently the area above the line I L must balance the vacant area below that line, or area A O N C, Fig. 177, equals area I K L, Fig. 176. That this is so is capable of mathematical proof.

So far we have made no mention of the back pressure due to expelling the exhaust water. As the velocity during

exhaust is the same at each point of the stroke as the velocity during the working stroke, the back pressure will be represented by the ordinates of the diagram Fig. 176, added to the ordinates of the diagram Fig. 177, and the combined area A Q S, Fig. 179, represents the energy lost on this account. Since the area A O N C is equal to the area I K L, the area A C Q S is evidently equal to twice area I K L.

The only condition which remains to be investigated is when a length of pipe l of diameter d intervenes between the valve opening and the cylinder, or when a similar pipe of any length and diameter is attached to the exhaust outlet. It is evident that the velocity of the water in the pipe must be dependent upon the velocity v of the piston at every point of the stroke. Since $Q = Av$, and the areas of the pipes vary as the diameters squared, the velocity v_1 in the pipe varies in relation to v inversely as the squares of the diameter d of the pipe and D of the cylinder, or—

$$v_1 : v :: D^2 : d^2$$

$$\therefore \quad v_1 = \frac{vD^2}{d^2} \qquad - \quad - \quad - \quad (8)$$

Similarly the weight w_1 of the water in unit length of the pipe varies in relation to w the weight of unit length of water in the cylinder directly as their diameters squared, or—

$$w_1 : w :: d^2 : D^2$$

$$\therefore \quad w_1 = \frac{wd^2}{D^2} \qquad - \quad - \quad - \quad (9)$$

As v is a function of v_c we may substitute the values v_1 and w_1 for v and w in equation (5), when—

$$p_c = \pm \frac{d^2}{D^2} \cdot \frac{w}{g} \cdot \left(\frac{D^2}{d^2} v_c\right)^2 \cdot \frac{1}{R}$$

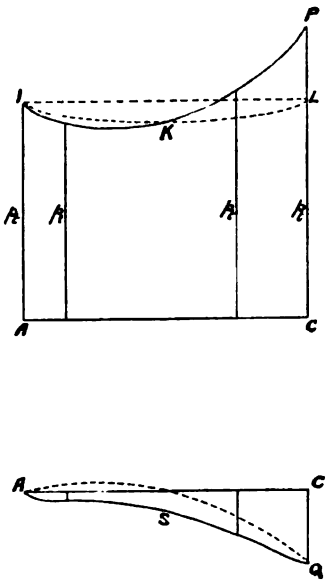
or

$$p_c = \pm \frac{d^2}{D^2} \cdot \frac{w}{g} \cdot \frac{v_c^2}{R} \qquad - \quad - \quad - \quad (10)$$

and w being taken equal to $l \times .434$, p_c will represent the accelerating force per square inch of piston area at the

commencement of the stroke. As w is a constant quantity throughout the stroke, the diagram of accelerating forces will be bounded by a straight line curve, as M O N in Fig. 177. The diagram so produced must now be subtracted from the diagram A I K P C, Fig. 178, to obtain the true values of p_t for these altered conditions.

A similar investigation may be made for the exhaust pipe, and the diagram, so produced, added to the diagram A C Q S, Fig. 179, taking care to observe the + and - signs.



Figs. 178 and 179.

We do not propose to investigate the energy lost by friction of pipes and bends, as the matter has already been treated in Chapter II., and the formulæ there given may be applied, so that we are now in a position to examine some of the leading designs of hydraulic engines, and note to what extent it has been found advisable to follow the precise arrangement of details demanded by our preliminary investigation.

Fig. 180 shows a sectional elevation of a Brotherhood engine, while Fig. 181 shows a cross section of the same. The design consists essentially of three cylinders A B C, fitted with single acting rams or pistons D E F, and placed at 120° to each other. These three pistons operate by means of connecting rods one common crank-pin G, which imparts circular motion to the shaft H. The pressure water is

admitted from the supply pipe I to each cylinder during the outward stroke of its piston by means of a revolving valve K, which is driven by the plate L attached to the end of the crank-pin G. The valve K is of simple construction, having a passage M ending in a splayed mouth of such dimensions that communication between the supply pipe I and cylinder port M[1] is maintained through 180°, or a half revolution of the valve. The alternate half of the valve is cut away, so as to allow free escape of the exhaust water during the other

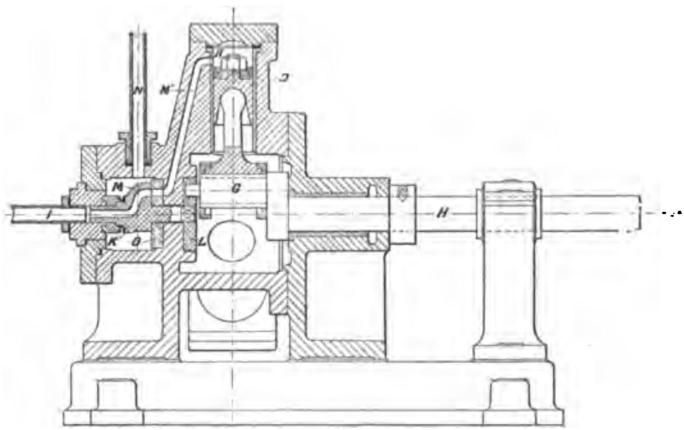


Fig. 180.

half revolution, thus permitting the exhaust water to flow away by the pipe N. The port face O is composed of lignum vitæ. Each piston is operated upon by pressure water during 180° of the revolution of the crank-pin, and, as there are three cylinders, there is no dead centre; the turning moment applied to the crank-shaft is, moreover, almost uniform, as will be seen by reference to the polar diagram, Fig. 182, in which vectors such as O A, O B, O E, O F represent the turning moments for the corresponding positions of the

crank-pin. These vectors also represent the velocity of flow in the supply pipe I, which is also very uniform.

In our preliminary examination we took the supply as being close to the cylinder, which we now see was justifiable, as the only water which is at rest at the ends of the stroke is the small quantity contained in the port M^1 and the cylinder clearance space. The flow in the pipe I may be rendered practically uniform by placing an air bell or a shock valve similar to Fig. 69, of suitable size, as close as possible to the

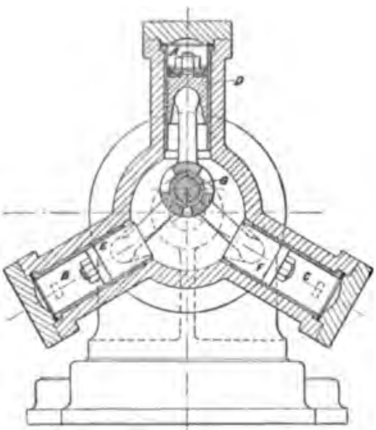


Fig. 181.

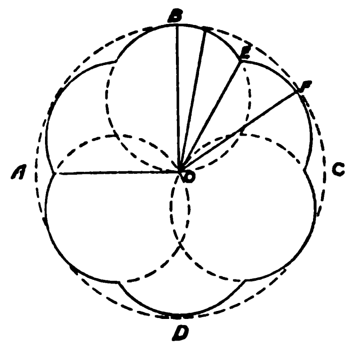


Fig. 182.

valve. It may be pointed out that the piston stroke is very short, thus allowing a moderately high number of revolutions per minute without excessive velocity of the entering and exhaust water—a condition tending, as we have seen, to improve the efficiency of the engine. In the figure the port M^1 is shown entering the cylinder with an abrupt enlargement, thereby causing loss by eddy currents; but owing to the high pressure (700 lbs. per square inch and upwards) usually applied to these engines, and the comparatively low velocity

of the entering water, the loss so caused forms a very small percentage of the whole energy imparted to the engine. Of course with low pressure, and the same velocity of entry, the losses are of moment, and the conditions laid down in our preliminary examination require to be rigidly adhered to if

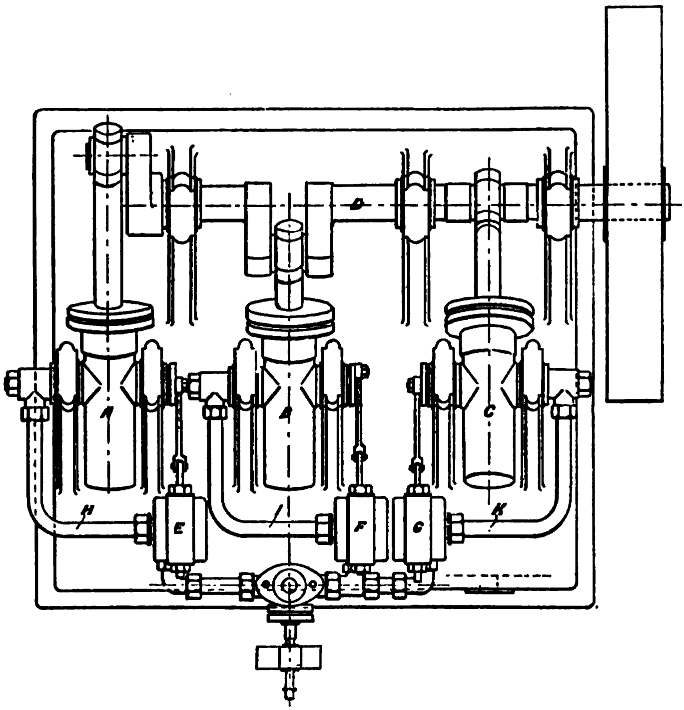


Fig. 183.

the engine is expected to show a satisfactory efficiency. The energy lost per square inch of piston area from this cause may be ascertained by an application of equations (3*a*) and (10). Equation (3*a*) must of course be multiplied by $\frac{d^2}{D^2}$,

if v be taken as the piston velocity as before, when the resulting diagram gives the losses per square inch of piston area. In applying equation (10) the second half of the diagram will evidently disappear, causing the energy and pressures represented by the first half to be lost or subtracted from the diagram of work.

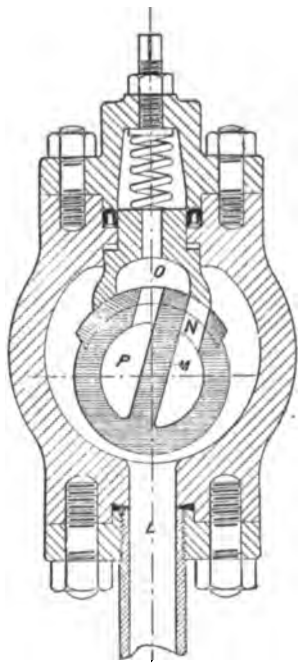


Fig. 184.

Fig. 183 shows in plan the general arrangement of a large size Armstrong engine in which three oscillating cylinders A B C are used, placed side by side, and operating a three-throw crank shaft D, having the cranks placed at 120° to each other, so that the turning moment applied to the crank shaft is precisely similar to that already described in reference to the Brotherhood engine. The valves E F G controlling the admission of water to the cylinders are of the reciprocating type, and are operated by connecting links worked from oscillating studs on the gudgeons of the cylinders. The water passes from the valves by pipes H I K connected to the gudgeons by a swivel union of the type shown in Fig. 62, and so through ports in the gudgeons to the cylinders.

In the smaller size of these engines, instead of the three reciprocating valves E F G, each cylinder is fitted with a valve of the type shown in Fig. 184, in which the oscillation of the cylinder operates the valve. The pressure water is admitted through the pipe L, and the oscillation of the

cylinder causes the oscillation of the valve M attached to the cylinder gudgeon, thus opening the port N to the pressure water L. The port N communicates through a port in the gudgeon with the cylinder, thus allowing pressure water to enter the cylinder so long as the port N is open. During the progress of the stroke the oscillation of the piston, and consequently that of the valve M, is reversed, so that when the piston is fully out the valve has again closed, and occupies the position shown in the figure. The further oscilla-

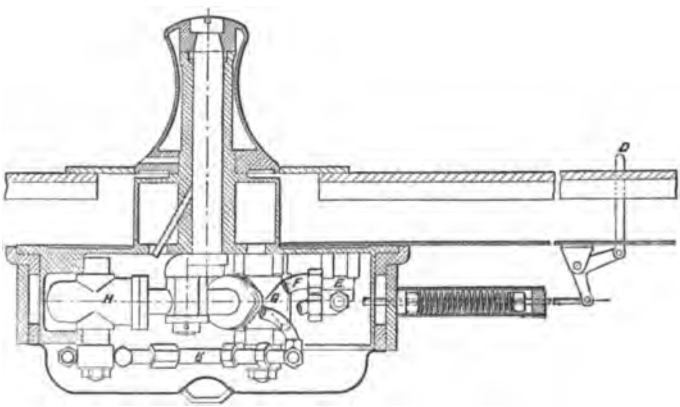


Fig. 185.

tion of the cylinder will cause the port N to open to the space O, which is in direct communication with the exhaust P. Valves of this type are liable to cause serious frictional losses due to the throttling of the water, as the valve, when nearly shut, is in the form of a long narrow slit.

Figs. 185 and 186 show in elevation and plan a three-cylinder Armstrong capstan, fitted with valves A B C of the type just described. The capstan is started in the usual manner by the button D, arranged in the floor, being depressed by the operator's foot, thus allowing water to pass

through the valve E to the supply pipe F, feeding the cylinders G H I. The exhaust water is conducted away by the waste pipe K.

The valve shown in Fig. 187 is interesting, as being the type used on the early designs of Armstrong engines. In these engines the pressure water was allowed to act on both sides of the piston during the outstroke. The piston rod was made of such diameter that its area was half of that of the cylinder, so that the piston was pushed outwards with a

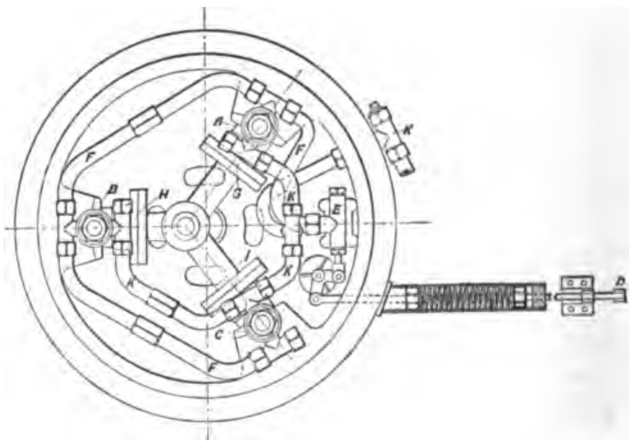


Fig. 186.

total pressure due to half its area, and the water contained in the forward end of the cylinder was returned to the supply pipe. On completing the outstroke the tail end of the cylinder was connected to the exhaust, while the forward end still communicated with the pressure supply. Thus the pressure water acting on the small area of the front of the piston drove back the piston, expelling the water from the large side to exhaust. By this means only half the work was done on the outstroke, the remaining half being performed on the return stroke. The cylinders were of the oscillating

type, and the valve A was formed solid with the gudgeon. The port B connects to the large side of the piston, and the port C to the small side. The port C is always open to the pressure supply D, while the port B is alternately open to the pressure supply D and exhaust pipe E. A small shock valve F was applied as shown to prevent the pressure in the

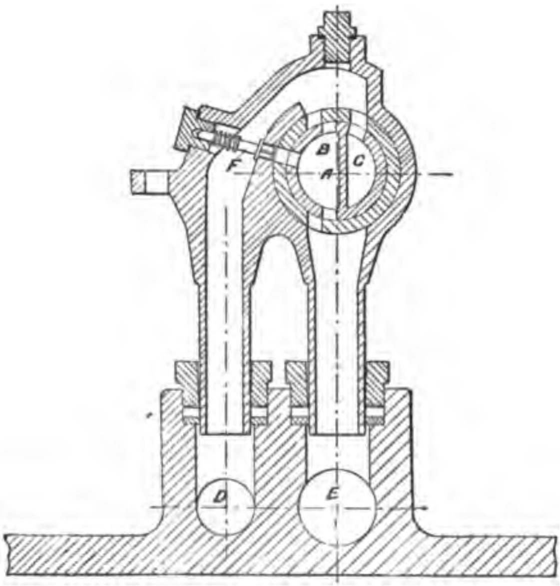


Fig. 187.

cylinder rising above that in the supply pipe, in case of any irregularity in the action of the valve.

In all the types of engine we have described up to the present no attempt has been made to economise water when working a light load. Several more or less successful attempts have been made to produce an engine which shall consume pressure water in some proportion to the useful

load. The best known of these engines is the revolving engine of Rigg.

Fig. 188 shows a sectional elevation of Rigg's engine. The design consists essentially of three or four cylinders such as A B C arranged radially about a pin or gudgeon D. Each cylinder is fitted with a piston or ram E F G, which is attached at its outer end to a revolving fly-wheel H by the

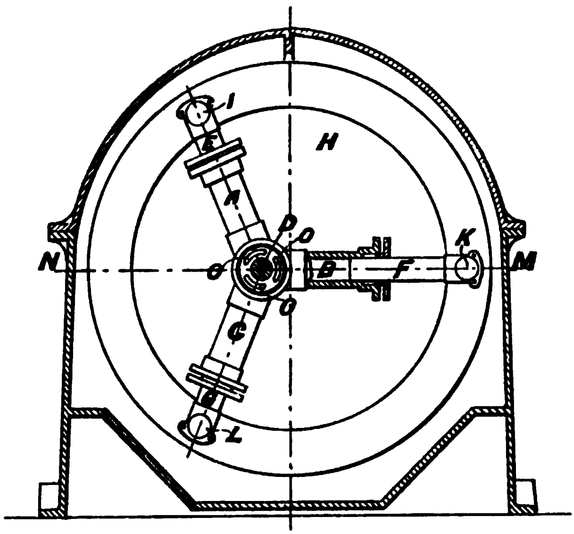


Fig. 188.

joints I K L. Now, if the axis of the fly-wheel coincides with the centre of the gudgeon D, it is evident that the cylinders and rams will be revolved about the gudgeon when the fly-wheel H is turned, but the rams will not make a reciprocating stroke in the cylinders. If now the centre of the gudgeon D is moved off the axis of the wheel H, as shown in the figure, each ram on arriving at M will project some distance out of the piston, while at N the ram will recede into the

piston. Thus in one complete revolution of the wheel H each ram will evidently make an out and return stroke, the length of this stroke being twice the eccentricity of the gudgeon D from the axis of the wheel H. If water pressure now be applied to each cylinder when at N, and the port opened to exhaust at M, the ram will be driven outwards, causing revolution of the wheel H and consequent revolution of the pistons and cylinders—hence the name revolving engine.

The quantity of water used is directly proportional to the length of piston stroke, and consequently to the eccentricity of the gudgeon D which corresponds to the crank throw in an ordinary engine. By shifting the gudgeon D nearer to, or farther from, the axis of H, the power of the engine is varied and also the consumption of water. The pressure water enters through the ports O operated by a valve of the type shown in Fig. 180. The gudgeon D, which is subject to all the conditions of stress of an ordinary crank-pin, has to be capable of adjustment in position whilst the engine is running. Fig. 189 shows the relay engine for controlling the gudgeon D. The gudgeon is securely attached to the two-ended ram P which passes into the cylinders Q R. By means of the internal plunger S the effective area of the ram P in the cylinder R is reduced to about half of its area in the cylinder Q. The cylinder R is always open to the pressure supply, while the cylinder Q is capable of communication to the pressure supply or exhaust by means of two small valves at T operated by a centrifugal governor not shown in the figure.

When the engine is running below its normal speed as in starting, or if overloaded, the governor operates a valve which allows the water in the cylinder Q to escape to exhaust, thus allowing the eccentricity of the gudgeon D to be increased, and consequently the power of the engine augmented. When the power of the engine is abreast of the load the governor will have acquired its normal position and closed the exhaust valve, thus locking the ram P in its new position.

If the load, or some part of it, be now removed, the engine will revolve quicker, thus causing the centrifugal governor to operate a valve connecting the cylinder Q to the cylinder R, and owing to the larger area of the ram P in the cylinder Q, the ram P will travel into the cylinder R, causing less eccentricity of the gudgeon D. When the speed of the engine again becomes normal the valve will be closed and the

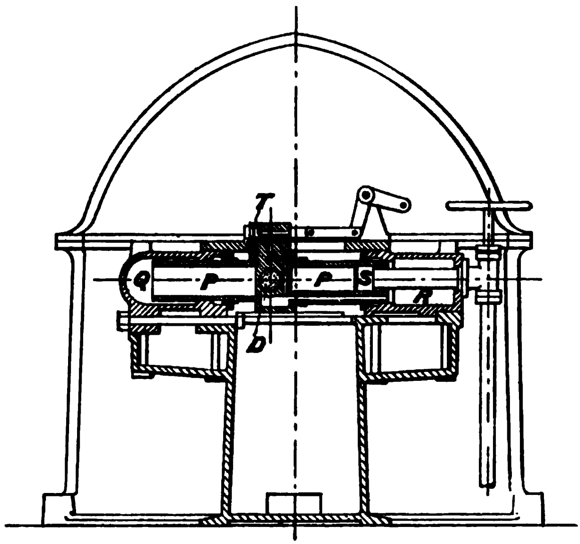


Fig. 189.

motion of the plunger P arrested. Thus the water consumption is automatically controlled according to the load applied to the engine.

There have been several attempts to attain this end, notably that of Hastie, who arranged for the crank-throw of an engine to be altered automatically by the variable turning moment required in the crank-shaft to overcome the load. A pair of hydraulic cylinders were arranged with plungers and

pitch chain connections to the crank-shaft, so that the load caused the chains to be partly wound round the shaft, thus driving the plungers back into their cylinders against the pressure water. The shaft was of cam form at the parts where the chains operated, so that at some point the turning moment required to overcome the load balanced the water pressure in the cylinders. This apparatus governed the crank throw of the engine, thereby producing economy of pressure water. The arrangement, though ingenious, has now dropped out of use.

The efficiency of hydraulic engines varies from about 50 to 80 per cent. For well-designed engines of the types illustrated, and working with a pressure of 700 lbs. per square inch and upwards, an efficiency of 70 to 80 per cent. may be expected. The horse-power is then given by the equation—

$$H.P. = \frac{pAL}{33000}.Rn \times C$$

in which p = pressure in pounds per square inch.
A = area of piston in square inches.
L = stroke in feet.
R = revolutions per minute.
n = number of single-acting cylinders.
C = efficiency – .70 to .80.

CHAPTER XXIV.

RECENT ACHIEVEMENTS.

Hydraulic Lifts.—Probably the most powerful combination of hydraulic lifts is that employed in connection with the hydraulic dock at the Union Iron Works, San Francisco, which is capable of raising a ship of 4,000 tons weight a height of 32 feet. Eighteen hydraulic rams are arranged on each side of the dock, which consists of a platform built of cross and longitudinal steel girders 62 feet wide, 440 feet long, provided with keel and sliding bilge blocks for the ship to rest upon. A set of four single-acting hydraulic plunger pumps 3½ inches diameter and 36-inch stroke, working at forty double strokes per minute, transmit water at a pressure of 1,100 lbs. per square inch to the thirty-six hydraulic rams, each of 30 inches diameter, with a stroke of 16 feet. On the top of each hydraulic ram is a 6-foot pulley over which eight steel cables 2 inches in diameter pass, one end of each cable being anchored to the bed plates supporting the cylinders, while the other is secured to the side girders of the platform. The illustration of the dock in Fig. 190 is from *Cassier's Magazine*, and shows a vessel in position on the platform.

When a ship is being lifted it sometimes happens that the load is not evenly distributed on the platform. Some rams, therefore, may carry a full load, while others are much underloaded. The platform is kept level by means of specially designed valve gear operated by the moving rams in such a manner that when one ram has a light load it moves ahead of the others, but in doing so lifts a lever and closes its inlet valve, so that the rams are practically stopping and

Fig. 190.—Hydraulic Dock at San Francisco.

[To face p. 338.

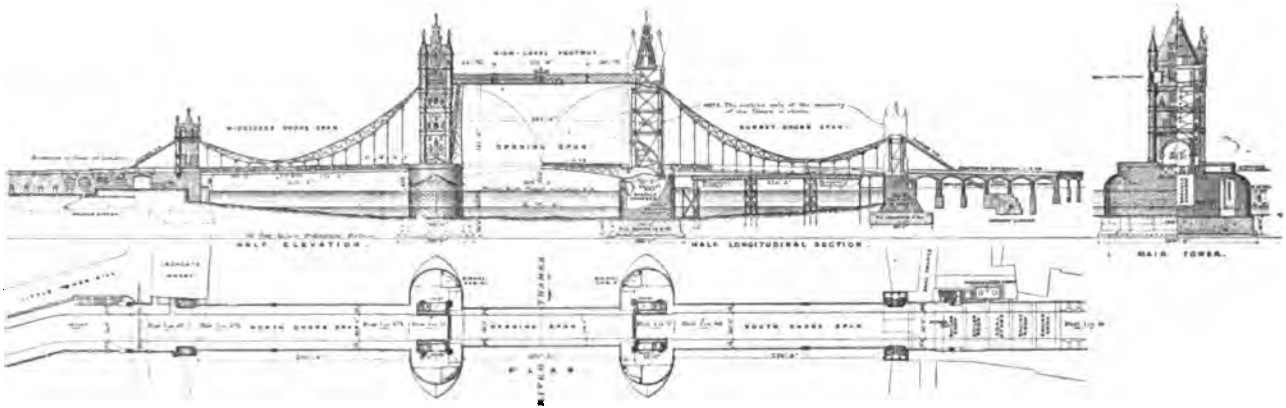
starting dependent upon the load which may come upon them, the valves being opened and closed automatically by the movement of each of the rams. The valve box is secured on the ram itself and moves up and down with it, the inlet and outlet pipes working through stuffing boxes in the usual manner.

The application of hydraulic power for effecting the opening of the bascules of the Tower Bridge over the Thames at London is shown in Figs. 191, 192, 193, and 194.

On each of the outside main moving girders quadrants are arranged having toothed racks bolted thereon. Two racks are placed on each quadrant, the pitch being 5.9 inches, and pinions mounted on two shafts across the bridge gearing to these quadrants. The lower shaft with its pinion is driven from the east end of the pier, and the upper one from the west on the south pier, while on the north pier the lower shaft is driven from the west, and the upper one from the east end. These pinions are actuated from gearing having a ratio of 6 to 1 by hydraulic engines placed in chambers at the ends of the piers, the machinery at each end of each pier being sufficient for the full requirements of one bascule, that at the other end of the pier being in reserve.

Each set of machinery consists of two three-cylinder hydraulic engines of unequal power, having pinions on their crank-shafts which gear into spur-wheels on an intermediate shaft, a pinion on which gears into the spur-wheel on the end of the rack-pinion shaft. The hydraulic engines were made of unequal power as a provision against the effect of wind on the large exposed surfaces of the bascules. It has been found, however, that it is not necessary to use more power than that given by one small engine.

The engines have three plungers 8½ inches diameter, 27-inch stroke in the large engines, and 7½ inches diameter and 24-inch stroke in the small engines. Each cylinder is provided with a separate working valve, separate spindles



Figs. 191 and 192.

being employed for the admission of pressure and the release of exhaust water. On the crank-shaft of each engine there is a brake wheel against which brake blocks attached to levers are thrust. The blocks are kept apart by hydraulic cylinders, and rams placed between the levers. They are drawn together by wire ropes and counter-weights when the bascules are standing. Before starting the bascules, pressure is admitted to the cylinders by releasing the brakes.

In ordinary working each bascule is raised and lowered by one hydraulic engine, the other three engines being in gear and running idle, the water circulating through their cylinders and valves. This provision is arranged in order that the power may be varied or the engine changed by the driver without having to leave his cabin. Clutches are provided by which any of the hydraulic engines can be thrown entirely out of gear. The time occupied in raising and lowering the bascule is about 1½ minutes.

At each end of each pier an accumulator is provided with a ram 22 inches in diameter and an 18-foot stroke. In the machinery chambers other hydraulic pumps are provided for delivering water to the top of the main towers for fire and domestic use. Provision is made for two hydraulic hoists having cradles 14 feet 9 inches long, 6 feet 6 inches wide, 11 feet high, the length of the lift being about 110 feet, for taking passengers to and from the high-level footways while the bascules are raised. The cradles are lifted and lowered by wire ropes from vertical cylinders, and rams placed in duplicate in the towers, safety gear being provided for gripping the guides and supporting the cradles in case of failure. Inter-locking gear is also arranged upon the cradles to prevent the hoist being started until both inside and outside doors are closed, or to prevent the doors being opened until the proper platform is reached and the hoist stopped.

The hydraulic power for the bridge is generated by two

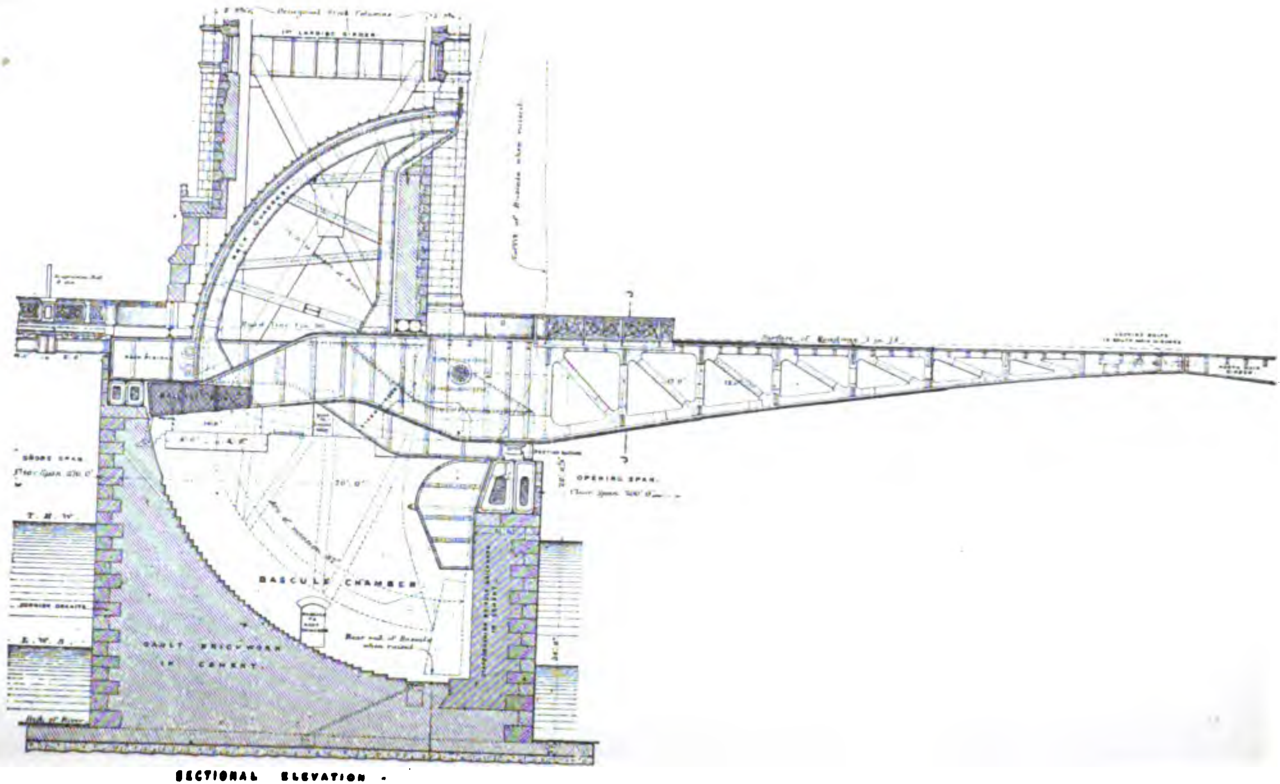


Fig. 193.

Fig. 194.—The Tower Bridge over the Thames.

[To face p. 342.

double tandem compound surface condensing engines, each of 360 I.H.P., the cylinders being 19¾ and 37 inches diameters respectively, while the pumps are 7¾ inches diameter and 38-inch stroke. One engine is sufficient to provide power for the bridge, while the other is held in reserve. The water pressure is 700 lbs. per square inch. The engines are supplied with steam by four Lancashire boilers, 7 feet 6 inches diameter, 30 feet long, working at 85 lbs. pressure. In addition to the four accumulators in the piers, there are at the engine-house two accumulators with rams 20 inches diameter having a stroke of 35 feet.

The pressure pipes are arranged in duplicate, while the return water pipes are single. The mains are protected from frost by hot-water pipes running alongside them, although a mixture of glycerine and water is employed in connection with the cylinders for the working of the bascules forming a small system of its own. Duplicate pumps actuated by hydraulic pressure placed within the south pier supply this subsidiary system. The machinery was designed by Sir W. G. Armstrong & Company in conjunction with and under the direction of Sir J. Wolfe Barry.

It has been found by experience that the time required for the bridge to be opened for the passage of vessels at any particular period is so short that it is found unnecessary to use the lifts for conveying passengers from the lower to the higher level. Pedestrians who wish to ascend to the upper footway can do so by means of 205 steps arranged within the towers.

The two bascules each weigh 1,070 tons, and they are carried on live ring rollers.

Water Balance Railways.—The author has introduced hydraulic brakes for controlling the motion of cars on cliff or inclined railways, using in connection with the water-balance system brakes which press against the rams under the influence of hydraulic pressure exerted through

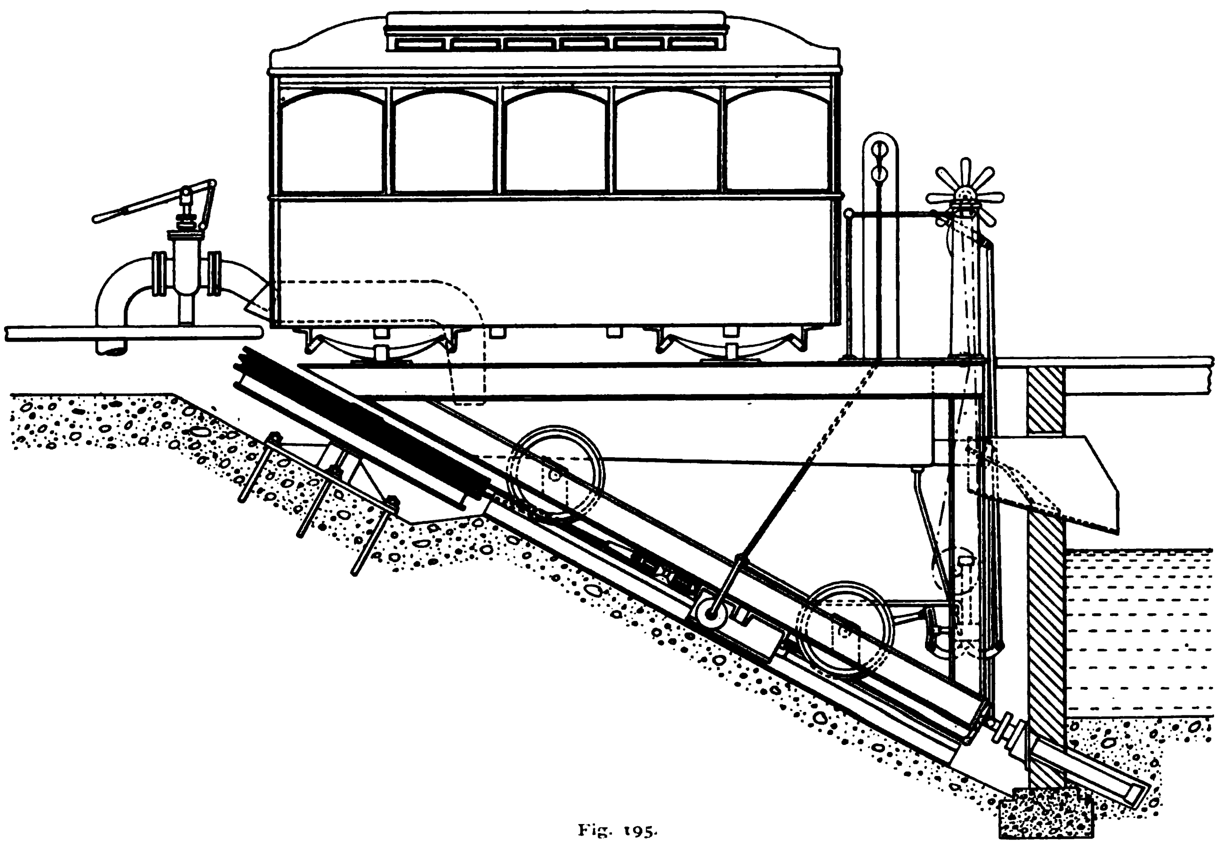


Fig. 195.

rams working in cylinders fed by pumps driven directly from one of the axles of the cars. Fig. 195 is an elevation of such a railway, similar to those constructed and erected by the author in various parts of the country; and Fig. 196 is a sectional elevation of the hydraulic rail-gripping brakes in use upon such cars.

The system of working in connection with these railways is to employ water in the form of ballast, which is introduced into the car when at the upper platform to overbalance the weight of the loaded car standing at the lower platform, and on the arrival of the water-ballasted car at the lower platform it discharges its water into a tank arranged there, from which

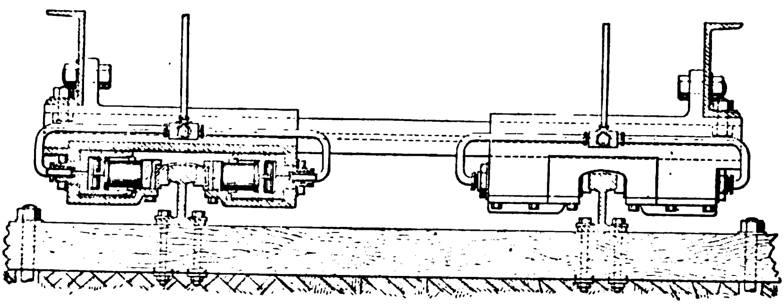


Fig. 196.

tank the water is again pumped back to a tank at the upper station, so as to enable the same water to be used over and over again.

The tanks are arranged between the girders or framework of the cars, and made of a capacity such as will contain water sufficient to overbalance the bottom car when fully loaded and the upper car having no passengers therein. The advantages of employing hydraulic balance as against hauling by direct driving of the top rope pulley is that the weight of the water introduced is regulated to suit the number of passengers to be carried, the conductor at the

lower station signalling to the brakesman at the top the number of passengers to be carried before the upper car is fully charged with water. It frequently happens that it is unnecessary to employ any water as ballast, owing to the preponderance of passengers for the down over those travelling on the up journey.

Hydraulic buffers are arranged at the lower platform, so that on the car striking one pair the water or liquid is driven out through a contracted passage into the cylinders of the opposite pair, thus forcing out the rams of the buffers ready for the journey of the next car down.

The arrangement of the hydraulic brakes for gripping the rails is shown in Fig. 196. The water or fluid under pressure acts behind rams which force outwards the slippers against the rail heads, the rail slippers being shaped to suit the head of the rail, and to thus grip it on the under side of the head, and prevent the car from mounting should any unforeseen contingency arise. A general arrangement of a cliff railway is shown in Fig. 197, one car being on the downward journey, the other in a correspondingly higher position on the other upward track, above the two bridges shown in the illustration, which is a photograph of the Lynton and Lynmouth Cliff Railway, which was constructed under the author's direction.

Glasgow Harbour Tunnel Lifts.—The hydraulic elevators employed in connection with the Glasgow Harbour Tunnel are more powerful than any yet constructed for a similar purpose, though the height of the lift is not so great as at the Eiffel Tower. The load at Glasgow on each cage is 12,000 lbs., and the maximum lift 72 feet. The Eiffel Tower lift was for 72 persons, and the height 420 feet.

There are six elevators in each shaft, three for raising and three for lowering vehicles. Fig. 198 illustrates three of the multiplying cylinders, and Fig. 199 shows the position of the car and the cylinders in the shaft. The diameter of the

Fig. 197.—Cliff Railway at Lynton.

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elevating cylinders is 13 inches, the lowering cylinders being 11½ inches diameter. The stroke of the rams in the cylinders is one-sixth of the car travel, the gearing being by three tandem sheaves, as shown in Fig. 198. The cylinders were tested to a pressure of 1,800 lbs. per square inch, the working pressure from the accumulators being 750 lbs. per square inch. The piston is 30 inches long, and the ram is 10 inches in diameter of cast iron; the piston, stuffing boxes, and gland being all of bronze.

The ram is 1¼ inches thick, and through the centre a 3-inch steel rod attached above the piston and passing through the head is arranged so as to rigidly connect the travelling sheave with the piston. The sheaves are respectively 52, 56, and 60 inches diameter.

Four steel lifting ropes are employed, the ends being attached to adjustment rods, two ropes passing down to each side of the cage. Each rope is ⅞ inch diameter, and is composed of six strands of steel wire wound round a hemp core, the strand itself consisting of eighteen wires, and each rope is tested up to 24 tons.

The main valves are bolted directly to the cylinder head. Each valve is 3 inches diameter, and has openings so graduated that the retarding or accelerating effort of the water when closing or opening the valve is constant. The levers from the operating gallery are connected with the pilot valve, which controls the operation of the main valve, so that the travel of the pilot valve lever is only slight in relation to the main valve. The main valve works on the differential principle, the area above being double that below the valve piston. The pressure is constantly below the valve piston, and the valve is moved down by admitting the pressure above the piston causing the valve to descend. The valve rises if communication is opened between the top of the valve cylinder and the discharge tank.

The lifting cylinders are arranged so that there is a preponderance of weight on the car side with pressure being

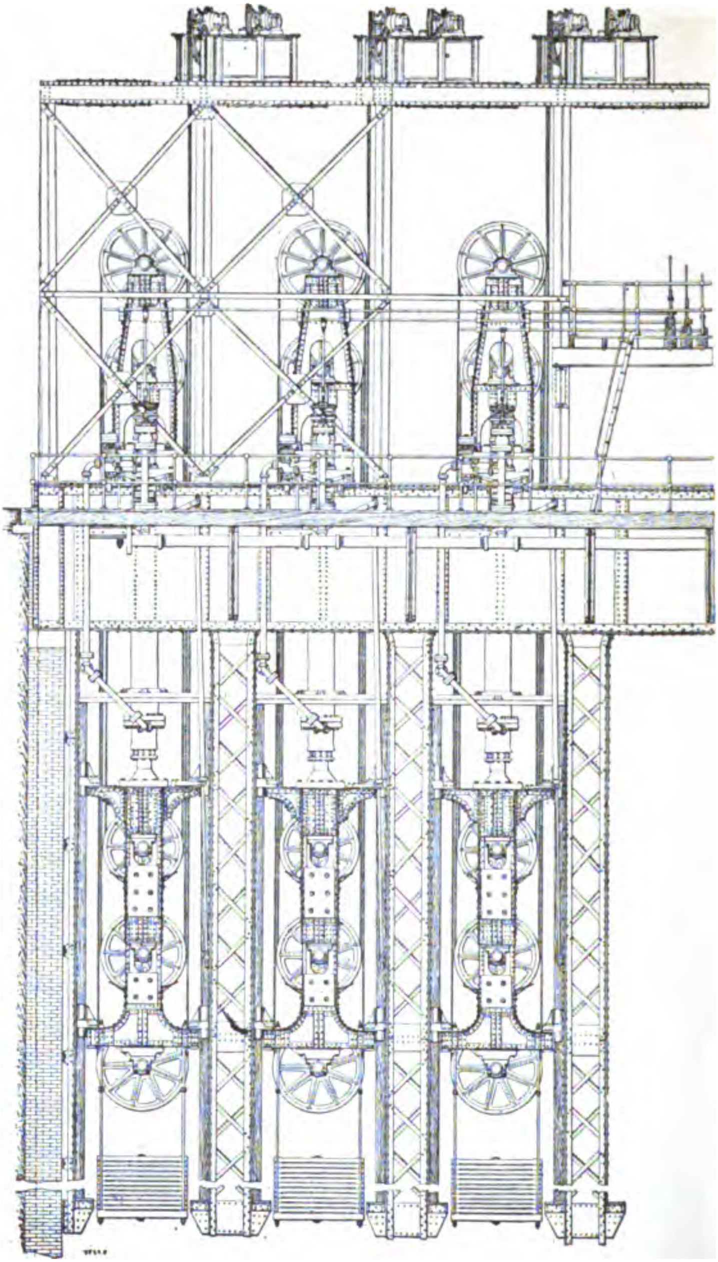

Fig. 198.

admitted above the piston to lift the load, or communication is established from above the piston to the discharge tank to lower the load. Water is consumed proportional to the load lifted, there being two powers. For load of 6,000 lbs. or less the cylinders use 37.8 gallons when lifting the load 74 feet, while for a greater load than 6,000 lbs., 70.7 gallons are consumed for the same travel. This change of power is rendered automatic by the use of a valve which remains closed with a load of less than 6,000 lbs., so that the water beneath the main piston lifts a balance check valve and is forced into a pipe connected with the main cylinder head. When lowering the car, however, this balance check valve closes and an unbalanced check valve lifts, thus opening communication from below the piston to the discharge tank. An amount of water equal in volume to the space beneath the piston is drawn in below the piston, and on the reverse stroke when lifting this water is introduced above the piston, so that the actual quantity of water used is that due to the displacement of the plunger only.

When lifting loads above 6,000 lbs. the preponderance of effort is below the piston of the automatic valve which rises and opens communication between the valve and the discharge tank.

The lowering cylinders are arranged so that the weight of the car is overbalanced and the tendency of the unloaded car would be to rise, but when loaded to descend thus using no water, the water in this case serving only as a brake. Should a vehicle be too light to overcome the overbalance and friction of the machine, water pressure is introduced into the cylinders.

Triple grip safety catches are fitted beneath each car arranged on the Otis Company's system, this Company having carried out and constructed the elevators.

Hydraulic Forging Press.—Fig. 200 is an illustration of a 4,000-ton hydraulic forging press in use at the works

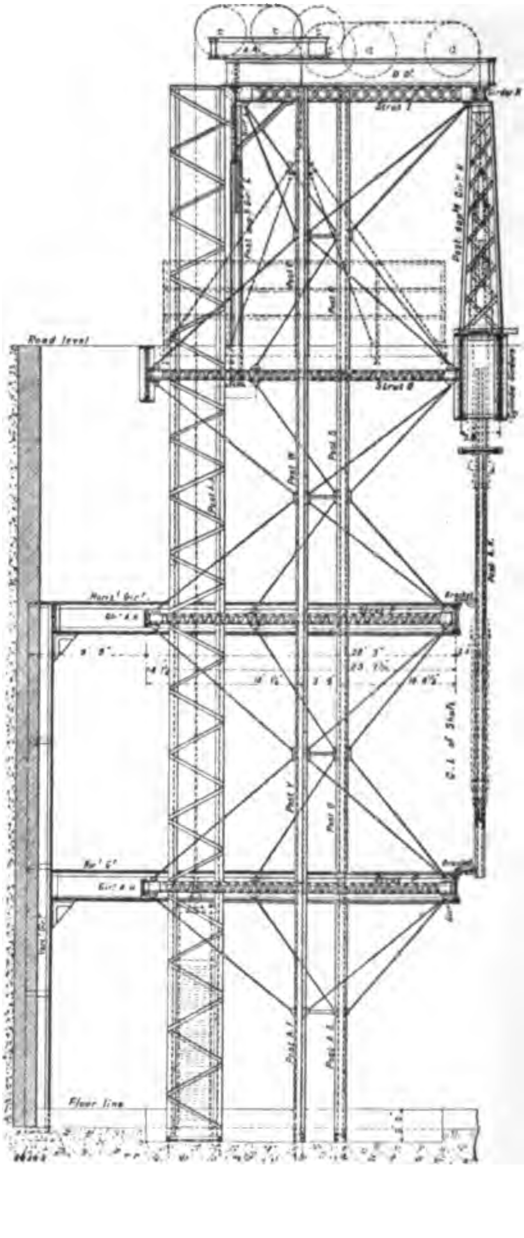

Fig. 199.

of Messrs Charles Cammell & Co. Ltd., Sheffield. This press, although not being by any means the largest of its kind in Sheffield, is probably one of the best examples of the heavy forging press now so generally adopted for dealing with massive forgings. Presses of over 10,000 tons power are working in the most satisfactory manner upon blocks of metal totally beyond the power of any steam hammer.

In the 4,000-ton Davy Press illustrated, which is from a photograph of Messrs Cammell's forge, two main rams of 36 inches diameter are mounted in the upper frame casting 9 feet 3 inches apart at the centres. Two lifting rams are also arranged thereon each of 9 inches diameter, the stroke of the press being 7 feet. The four columns carrying the head are of steel 20 inches diameter, the centres of the same being 15 feet in one direction, and 6 feet 4 inches in the other. The distance between head and block is 21 feet. The press is supplied with water at 4,500 lbs. per square inch pressure, by means of three single-acting pump plungers, each 6 inches diameter and 12-inch stroke, driven from the crank-shaft of a pair of steam engines having cylinders of 34 inches diameter. The supply water to the pumps is fed from a low-pressure main at 60 lbs. pressure, this low pressure being also useful in filling the main cylinders when the smaller or lifting rams are working, raising the crosshead and tool. This arrangement for supplying pressure water to the pump barrels admits of small valves being fitted to the pumps.

The pumps work at varying speeds up to sixty or more revolutions per minute, the speed of lifting when the low-pressure water is introduced into the main cylinders and the high pressure to the lifting cylinders being 8 inches per revolution, while the speed of descent under the full load is ½ inch per revolution, the relative areas of the lifting and lowering rams being 16 to 1. Two levers control the whole movements of the press, one of these being also for starting the pumps. In operation the forging tool is

raised 2 feet per second, this quick motion being necessary to admit of moving the forging readily while hot.

The employment of the forging press admits of a much lower building being constructed than would be possible with a steam hammer. This advantage also enables cranes to travel over the entire press, and thus to command the whole forge area. The two travellers shown in Messrs Cammell's forge are respectively of 150 and 110 tons lifting power.

Niagara Power.—The amount of water power flowing to waste, so far as mechanical energy is concerned, in various parts of the world, is truly appalling in its immensity. The installations, however, at Tivoli (by means of which power developed there is transmitted to Rome, 16 miles distant), at Geneva, Schaffhausen, Zurich, Telluride (in Colorado), and other places, are amply sufficient to warrant the assertion that the trend of commercial utilisation of such water aste is becoming a factor for profitable consideration wherever mechanical power of any kind for any purpose is required.

The flow of water at the crest of the Horse Shoe Falls at Niagara has been found to be about 275,000 cubic feet per second, and it has been estimated that over 100,000,000 tons per hour pass over the Fall. The plunge of this immense mass of water from one level to another of 165 feet has enabled the Fall to be harnessed, and energy taken therefrom by the Niagara Falls Power Company.

The theoretical horse-power which is available at the Falls has been given by the United States Government engineers as 6,750,000 H.P., an amount which, if produced by steam, would necessitate the consumption of more coal than is at present raised throughout the world.

The illustration shown in Fig. 201, by kind permission of Messrs Cassier, gives a bird's-eye view and section of the Niagara installation, from which it will be seen that water is taken from the upper level above the first Fall, and allowed

Fig. 200.—4000-Ton Hydraulic Forging

PRESS (Cammell's Works, Sheffield).

[*To face p.* 352.

to pass through turbines mounted in a power house and wheel pit, the discharge or tail-race water from the turbines passing through the tunnel leading out into the lower level below the Falls. The wheel pit of the Niagara Falls Power Company is a long slot cut in the rock, instead of a group of small wheel pits, and the tail-race from each wheel or turbine is connected by a short curve to the main tail-race tunnel.

The turbines are arranged, some for developing 1,100 H.P. per wheel, others 5,000 H.P. per wheel. The 1,100 H.P. turbines are of the Jonval type, the fall of water being 140 feet on to the wheels, which make 250 revolutions per minute.

Various manufacturing establishments have already erected machinery on the ground near to the Niagara Falls installation. But beyond the mere local uses for the power, and the enormous development of industries which must attend this form of producing mechanical energy from one centre, other applications are being made for transmitting the power to a distance, for the purpose of displacing private plants at present employed for electric lighting and for ordinary manufacturing purposes.

Seeing that the transmission of oil by means of a pipe line a distance of over 400 miles, and also the transmission of natural gas by a pipe line a distance of 120 miles, have been found feasible, it is not too much to expect that ere long there will be distributed mechanical power to similar distances, and with results which will be not only economical but advantageous alike to the users and the districts where it is employed, by reason of its displacing private steam or other power-generating motors and leaving the atmosphere free from the products of combustion necessarily attendant upon the use of coal for such power-producing purposes.

Turbine for Small Fall.—As an example of what is possible under difficult and unpromising conditions, the

turbine installed at Strensham Mills, near Worcester, is worthy of notice. Owing to the natural conditions of the River Avon the water head available varies from 4 feet in summer, with a diminished supply, to 2 feet in winter, with an excessive supply.

The horse power required was 40, and it became necessary to design a turbine adapted to the varying conditions. A Jonval turbine was selected, having a double ring of vanes. The outer ring of vanes is sufficient to supply the power under a 3 foot head, and as the head is diminished by flood, the gates closing the inner ring of guide passages are opened to allow a larger quantity of water to pass.

The method of using two rings of vanes allows scope in designing, as the outer vanes can be speeded for correct working under a 3 foot head, and the inner for correct working under a 2 foot head.

The turbine is 13 ft. 2 in. in diameter, and makes 14 revolutions per minute.

Fig. 201.—Bird's-eye View of Hydraulic Power Installation at Niagara Falls. [*To face p.* 354.

APPENDIX.

TABLE XII.—Pressure of Water.*

Showing pressure of water in pounds per square inch for every foot in height to 270 feet. By this Table, from the pounds pressure per square inch the feet head is readily obtained, and *vice versâ.*

Feet Head.	Pressure per square inch.	Feet Head.	Pressure per square inch.	Feet Head.	Pressure per square inch.	Feet Head.	Pressure per square inch.	Feet Head.	Pressure per square inch.	Feet Head.	Pressure per square inch.
1	0.43	46	19.92	91	39.42	136	58.91	181	78.40	226	97.90
2	0.86	47	20.35	92	39.85	137	59.34	182	78.84	227	98.33
3	1.30	48	20.79	93	40.28	138	59.77	183	79.27	228	98.76
4	1.73	49	21.22	94	40.72	139	60.21	184	79.70	229	99.20
5	2.16	50	21.65	95	41.15	140	60.64	185	80.14	230	99.63
6	2.59	51	22.09	96	41.58	141	61.07	186	80.57	231	100.06
7	3.03	52	22.52	97	42.01	142	61.51	187	81.00	232	100.49
8	3.46	53	22.95	98	42.45	143	61.94	188	81.43	233	100.93
9	3.89	54	23.39	99	42.88	144	62.37	189	81.87	234	101.36
10	4.33	55	23.82	100	43.31	145	62.81	190	82.30	235	101.79
11	4.76	56	24.26	101	43.75	146	63.24	191	82.73	236	102.23
12	5.20	57	24.69	102	44.18	147	63.67	192	83.17	237	102.66
13	5.63	58	25.12	103	44.61	148	64.10	193	83.60	238	103.09
14	6.06	59	25.55	104	45.05	149	64.54	194	84.03	239	103.53
15	6.49	60	25.99	105	45.48	150	64.97	195	84.47	240	103.96
16	6.93	61	26.42	106	45.91	151	65.49	196	84.90	241	104.39
17	7.36	62	26.85	107	46.34	152	65.84	197	85.33	242	104.83
18	7.79	63	27.29	108	46.78	153	66.27	198	85.76	243	105.26
19	8.22	64	27.72	109	47.21	154	66.70	199	86.20	244	105.69
20	8.66	65	28.15	110	47.64	155	67.14	200	86.63	245	106.13
21	9.09	66	28.58	111	48.08	156	67.57	201	87.07	246	106.56
22	9.53	67	29.02	112	48.51	157	68.00	202	87.50	247	106.99
23	9.96	68	29.45	113	48.94	158	68.43	203	87.93	248	107.43
24	10.39	69	29.88	114	49.38	159	68.87	204	88.36	249	107.86
25	10.82	70	30.32	115	49.81	160	69.31	205	88.80	250	108.29
26	11.26	71	30.75	116	50.24	161	69.74	206	89.23	251	108.73
27	11.69	72	31.18	117	50.68	162	70.17	207	89.66	252	109.16
28	12.12	73	31.62	118	51.11	163	70.61	208	90.10	253	109.59
29	12.55	74	32.05	119	51.54	164	71.04	209	90.53	254	110.03
30	12.99	75	32.48	120	51.98	165	71.47	210	90.96	255	110.46
31	13.42	76	32.92	121	52.41	166	71.91	211	91.39	256	110.89
32	13.86	77	33.35	122	52.84	167	72.34	212	91.83	257	111.32
33	14.29	78	33.78	123	53.28	168	72.77	213	92.26	258	111.76
34	14.72	79	34.21	124	53.71	169	73.20	214	92.69	259	112.19
35	15.16	80	34.65	125	54.15	170	73.64	215	93.13	260	112.62
36	15.59	81	35.08	126	54.58	171	74.07	216	93.56	261	113.06
37	16.02	82	35.52	127	55.01	172	74.50	217	93.99	262	113.49
38	16.45	83	35.95	128	55.44	173	74.94	218	94.43	263	113.92
39	16.89	84	36.39	129	55.88	174	75.37	219	94.86	264	114.36
40	17.32	85	36.82	130	56.31	175	75.80	220	95.30	265	114.79
41	17.75	86	37.25	131	56.74	176	76.23	221	95.73	266	115.22
42	18.19	87	37.68	132	57.18	177	76.67	222	96.16	267	115.66
43	18.62	88	38.12	133	57.61	178	77.10	223	96.60	268	116.09
44	19.05	89	38.55	134	58.04	179	77.53	224	97.03	269	116.52
45	19.49	90	38.98	135	58.48	180	77.97	225	97.46	270	116.96

* For permission to quote the Tables given in this Appendix, the Author is indebted to the kind courtesy of the Worthington Pumping Engine Company.

TABLE XIII.—Action of Pumps: Diameters, Areas, and Displacements.

Diameter.	Area.	Displacement in Imperial Gallons per foot of Travel.	Diameter.	Area.	Displacement in Imperial Gallons per foot of Travel.	Diameter.	Area.	Displacement in Imperial Gallons per foot of Travel.
1/8	.0122	.0005	7 3/4	47.17	2.037	19 1/4	291.0	12.571
1/4	.0490	.0021	8	50.26	2.171	19 1/2	298.6	12.900
3/8	.1104	.0047	8 1/4	53.45	2.309	19 3/4	306.3	13.232
1/2	.1963	.0084	8 1/2	56.74	2.451	20	314.1	13.569
5/8	.3068	.0132	8 3/4	60.13	2.597	20 1/2	330.0	14.256
3/4	.4417	.0190	9	63.61	2.747	21	346.3	14.960
7/8	.6013	.0259	9 1/4	67.20	2.903	21 1/2	363.0	15.681
1	.7854	.0339	9 1/2	70.88	3.062	22	380.1	16.420
1 1/8	.9940	.0429	9 3/4	74.66	3.225	22 1/2	397.6	17.176
1 1/4	1.227	.0530	10	78.54	3.393	23	415.4	17.945
1 3/8	1.484	.0641	10 1/4	82.51	3.564	23 1/2	433.7	18.735
1 1/2	1.767	.0763	10 1/2	86.59	3.740	24	452.3	19.539
1 5/8	2.073	.0895	10 3/4	90.76	3.920	24 1/2	471.4	20.364
1 3/4	2.405	.1038	11	95.03	4.105	25	490.8	21.202
1 7/8	2.761	.1192	11 1/4	99.40	4.294	25 1/2	510.7	22.062
2	3.141	.1356	11 1/2	103.8	4.484	26	530.9	22.935
2 1/8	3.546	.1531	11 3/4	108.4	4.682	26 1/2	551.5	23.824
2 1/4	3.976	.1717	12	113.0	4.881	27	572.5	24.732
2 3/8	4.430	.1913	12 1/4	117.8	5.088	27 1/2	593.9	25.656
2 1/2	4.908	.2120	12 1/2	172.7	5.300	28	615.7	26.598
2 5/8	5.411	.2337	12 3/4	127.6	5.512	28 1/2	637.9	27.567
2 3/4	5.939	.2565	13	132.7	5.732	29	660.5	28.533
2 7/8	6.491	.2804	13 1/4	137.8	5.952	29 1/2	681.4	29.522
3	7.068	.3053	13 1/2	143.1	6.182	30	706.8	30.533
3 1/8	7.669	.3313	13 3/4	148.4	6.410	31	754.8	32.607
3 1/4	8.295	.3583	14	153.9	6.649	32	804.2	34.741
3 3/8	8.946	.3864	14 1/4	159.4	6.886	33	855.3	36.949
3 1/2	9.621	.4156	14 1/2	165.1	7.132	34	907.9	39.221
3 5/8	10.32	.4458	14 3/4	170.8	7.388	35	962.1	41.562
3 3/4	11.04	.4769	15	176.7	7.633	36	1017.9	43.973
3 7/8	11.79	.5193	15 1/4	182.6	7.888	37	1075.2	46.448
4	12.56	.5426	15 1/2	188.6	8.147	38	1134.1	48.993
4 1/4	14.18	.6125	15 3/4	194.8	8.415	39	1194.6	51.607
4 1/2	15.90	.6868	16	201.0	8.683	40	1256.6	54.259
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